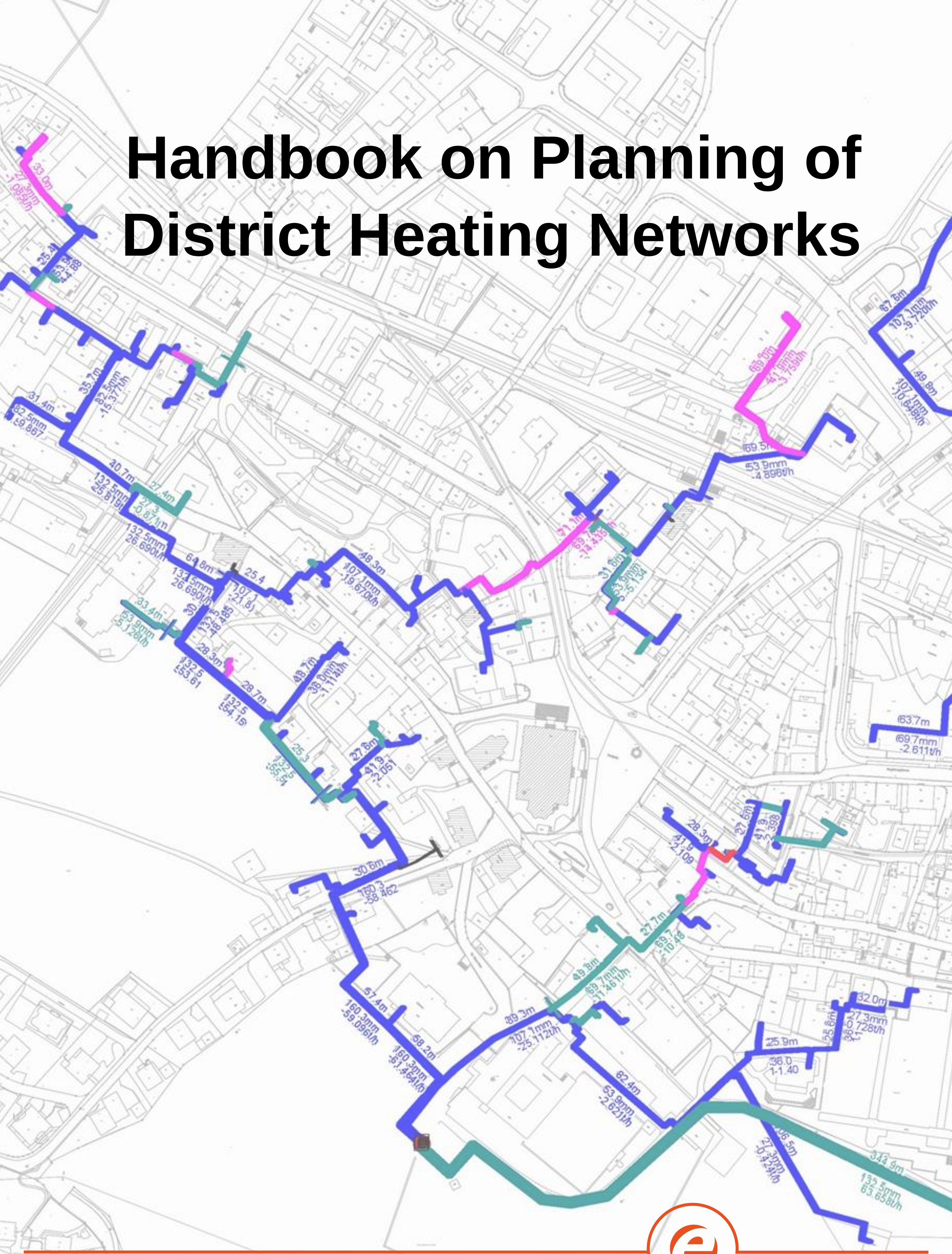
Handbook on Planning of District Heating Networks



QM District Heating

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Intention and Motivation

District heating enables the use of waste heat as well as the utilization of renewable energies. Therefore, it gains in relevance. The **Handbook on Planning of District Heating Networks** presents an introduction to technical and operational methods in order to realise a district heating network. It assits to organize and operate new district heating networks efficiently and under economic premises.

Target audience

- Central heating engineers and heating planners,
- Civil engineers and specialists within pipeline construction,
- Senior staff members within planning companies dedicated to heating, building techniques and excavation,
- Staff members of energy agencies,
- Operators of district heating networks.

Scope of work

The handbook describes basics in the fields planning, execution and optimisation of **Heat Distribution** and **Heat Exchange** within **District Heating Networks**. The descriptions rely to the following range of applications:

- Liquid water is used as heat carrier medium.
- The design reflects to directly used heat with temperatures above 40 °C.
- Heat distribution will be granted through pre-insulated jacket pipes that operate within a continuous operation temperature of up to 120 °C to 140 °C.

The distribution network design is presented none related to the technique of heat generation. Techniques of heat generation are described as well as the interaction between heat generation and distribution. Furthermore, pointed out are characteristics of wood energy, waste heat and ambient heat. The detailed description of heat generation is not the main focus of this handbook.

This handbook has been created on behalf of the Swiss Federal Office of Energy in the framework of the initiative "EnergieSchweiz".

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Content

The handbook consists of the following parts:

Part 1 Basics reflects the framing conditions, heat generation, heat distribution and heat exchange. It describes useful basic information required to an early project state.

Part 2 Planning and Calculation deals with the information required for execution planning. It deepens the topics introduced in chapter 1.

Part 3 Optimisation relates to district heating network analysis and the development of methods for optimisation.

As supplemental to the three core topics the **appendix** summarises design specifications as well as information to norms and literature.

Responsibility and use

The content of the handbook relies to the experience of the authors and the corresponding technical literature. In addition to this the editing has been accompanied by professional associations and industry representatives. Even though that the information has been improved in the best manner no liability can be granted. The handbook should be used as basis for education and further training and will be updated on a regular basis. Comments and suggestions for improvements are highly appreciated.

This handbook was originally written for the application in Switzerland and refers to Swiss laws and standards in several passages. For application in other countries, the corresponding laws and standards of the respective country must be applied.

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Basics

1 Introduction

1.1 Development of district heating

With an average annual temperature of around 8 °C and 200 to 300 heating days per year, the heating of buildings in our latitudes plays an important role. Heating and hot water are prerequisites for our comfort, but at the same time they are also responsible for a large part of the consumption of fossil fuels.

The principle of central heating was used with hypocausts for heat distribution by means of warm air in floor and wall heating systems as early as 2000 years BC in the Greek civilisation and was later also used by the Romans. Nevertheless, the time of open fireplaces and stoves in Europe lasted for many centuries. For example, closed fireplaces and stoves made of stone and clay appeared around 1000 AD. Tiled stoves were used from the 15th century onwards and iron stoves from the 18th century. Today's form of central heating only established itself after the Second World War. Until then, solid fuels were used almost exclusively. The use of oil only became wide-spread at the end of the 1940s and natural gas somewhat later.

The first district heating dates back to 1332, when water from hot springs was used to heat about 40 houses in Chaudes-Aigues in the French Massif Central [1]. However, the first district heating network in the modern sense is a steam-powered network built in Lockport (New York, USA) that was extended to several kilometres [2], [3]. From the 1920s and increasingly from 1960 onwards, the development of district heating networks began, espe-cially in northern countries. Figure 1.1 shows an example of a network in Leipzig in the 1980s. In Denmark, Swe-den, Finland, Iceland, Poland, the Czech Republic and Austria more than 20% of all residential buildings are con-nected to district heating today [3], [7], and [9].

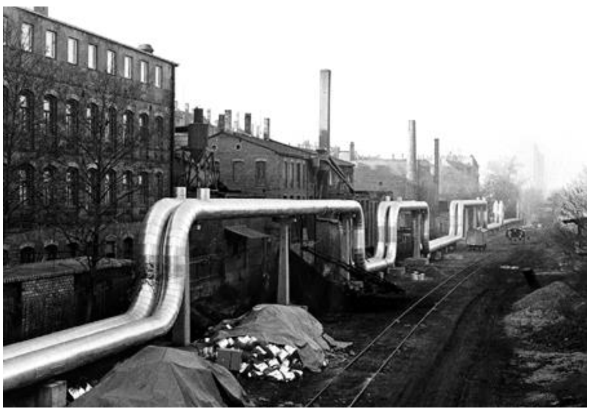


Figure 1.1 District Heating, Leipzig 1986, GDR [4].

1.2 Terms and Significance of District Heating in Switzerland

The "Energy Statistics of Switzerland" (Gesamtenergiestatistik der Schweiz) describes district heating:

"District heating consists of a heat supply system for which public space is used for the transport and distribution and which sells heat to third parties." [18].

Restrictions concerning the selling of heat and the use of public space have no connection to technical specifications. Therefore, this handbook applies even if the heat generator and heat consumer are organized within the same body and where no public land is required.

For smaller networks the term local heating or microgrid is used. For example, in Germany, this term describes the transfer of heat for heating and hot water between buildings with a capacity between 50 kW and a few mega-watts [10]. Switzerland (Minergie®) uses this term for the heat generation supplied to some buildings without a mandatory sale to third parties [11]. Because of the overlapping of microgrids and district heating, the term district heating is generally used in this handbook. In Germany, only the term district heating is considered to be technically and legally correct. A distinction between central heat generation and district heating does not depend on distance and size of the network [58]. Therefore, currently operated district heating networks cover a wide range of services with connected loads from less than 100 kW to 1 GW.

The statistically recorded figure for district heating is 18,300 TJ/a, which is about 2.2 % of Switzerland's energy consumption of about 840,000 TJ/a [18]. The data on district heating is based on statistical investigations carried out since 1978, which only covers the large networks supplied mostly by waste incineration plants. Smaller networks using wood, other renewable energies, heat pumps, oil and natural gas are therefore not included in the statistics, which is why the effective importance of district heating is significantly greater.

According to [19] there is a potential for district heating to supply a heat demand of about 17 TWh/a or 61,200 TJ/a by 2050, which can be covered by renewable energy sources. This corresponds to about 38 % of the long-term heat demand for space heating and hot water in Switzerland.

1.3 Temperature level

With increasing network temperature, the heat losses of the district heating network increase. If certain temperatures are exceeded, the demands on equipment and pipes also increase. The maximum supply temperature is therefore an important parameter for the differentiation of district heating networks. While space heat of around 40 °C can be used directly for new buildings, a minimum temperature of 60 °C is required for pre-heated domestic hot water in order to prevent legionella. For domestic hot water water therefore supply temperatures of approxi-mate-ly 70 °C are presupposed in the district heating network.

Apart from direct applications of heat, there is also an increasing interest in the distribution of heat at temperature levels of only 6 °C to 20 °C, for example for decentralised heat pumps. In the foreground is the use of waste heat and ambient heat. These networks are also called "anergy networks" or "cold district heating" (kalte Fernwärme) [16]. In order to reduce the primary energy input for covering an effective energy demand, the use of exergy (the proportion of energy used for work) must be reduced to a minimum. In heat distribution, this is made possible by reducing the temperature level with so called "LowEx district heating" [20].

If the distribution of water serves to transfer heat away from consumers, this is also referred to as "district cooling". Connecting heating and cooling consumers through networks with combined use offers additional potential for saving primary energy and is also referred to as "thermal area networking".

The majority of today's district heating networks are operated with supply temperatures of over 70 °C and are used for direct heating and usually also for the supply of hot water. The present handbook describes the necessary technology for the temperature range from 40 °C to 140 °C. The design limit corresponds to the continuous operating temperature of pre-insulated rigid steel pipes, which is usually guaranteed by pipe manufacturers. Since increased safety requirements must be complied with for temperatures above 110 °C, a distinction is also made between the following terms for heat distribution:

"Hot water" is used for heat distribution up to 110 °C.

"High temperature water" refers to water above 110 °C.

1.4 Pros and cons

District heating is becoming increasingly important as it provides a high level of comfort for consumers and enables the use of different heat sources. The heat sources for the supply temperatures of over 40 °C dealt with in this handbook are usually automatic wood-fired heating plants, waste heat from waste incineration plants or industry, ambient heat used with heat pumps and solar thermal energy.

For wood as an energy source, district heating enables the use of large, automatically operated heat generators. With the help of efficient fine dust separators and automated plant operation, it can be ensured that the increased use of wood for energy does not lead to more smog, as may occur with unsuitably operated small-scale furnaces.

A further advantage of district heating is the space gained in the building of the heat consumer, as an oil tank, chimney and firing system are no longer required.

In addition, district heating customers do not have to worry about expenses concerning fuel delivery, service and chimney sweeps.

The advantages of district heating are offset by the additional losses and costs for heat distribution. Since a district heating network requires high investments, the capital costs can account for more than 50% of the total costs [14]. In the case of wood, oil or natural gas, the fuel costs for covering the heat distribution losses are significant, while the electricity costs for the pump usually make up a smaller share [14].

With regard to the primary energy consumption, central heat generation can have further advantages depending on the application. For example, the use of otherwise unusable waste heat enables the saving of fossil fuels in decentralised heating systems. If there is no waste heat available, a district heating plant enables higher efficiencies to be achieved using larger heat generators which can be better adapted to the load condition. A wider range of energy sources can be used and combined. For example, a renewable base load coverage can be combined with a fossil-fuelled peak load boiler. The use of combined heat and power (CHP) for the base load share can also be made possible for wood.

The advantages of district heating do not always outweigh the disadvantages for certain applications. For exam-ple, district heating generated with natural gas is generally only advantageous compared to decentralised gas heating systems if electricity is generated with wood instead of gas and the waste heat is used as district heating.

1.5 Efficiency and costs

With a given connected load and supply temperature the efficiency and profitability of a district heating network are influenced by the following factors:

1. The **heat losses** of the network cause an additional heat demand. In the case of a boiler, more fuel is needed. The energy content corresponds to the heat losses divided by the boiler efficiency. In addition, the heat losses must be taken into account when dimensioning the heat generation and network. The heat losses increase with increasing surface area of the piping and thus with increasing pipe diameter. Heat loss decreases with improved thermal insulation.
2. A reduction in **pipe diameter** results in lower capital costs and lower fuel costs. At the same time the pressure loss and pumping capacity increase resulting in higher operating costs.
3. The **temperature difference** between flow and return flow influences the heat output transported at a given volumetric flow rate.

The following applies to the heat output:

$$\dot{Q} = \dot{m}\ c_p\ \Delta T = \dot{V}\ \rho\ c_p\ \Delta T = v\ A\ \rho\ c_p\ \Delta T$$

$\dot{Q}$ = Heat Capacity (kW)

$\dot{m}$ = Mass Flow Rate (kg/s)

$\dot{V}$ = Volumetric Current (m^3/s)

v = Flow Speed (m/s)

A = Pipe area (m^2)

ρ = Density of Water at 60 °C (983 kg/m^3)

c_p = Heat Capacity of Water at 60 °C (4.183 kJ/(kg K))

ΔT = Temperature Difference = (T_{SPLY} – T_{RTN}) in (K)

Figure 1.2 shows the volume flow required to transport 1 MW of heat output as a function of the temperature difference.

A high temperature difference allows the use of smaller pipes for a given connected load, which reduces capital costs and heat losses.

For a given network, an increase in the temperature difference enables an increase in the connected load. Con-versely, if the temperature difference is not reached, the network output is reduced. For this reason, it must be ensured that the heat transfer stations are operated correctly and achieve the required temperature difference.

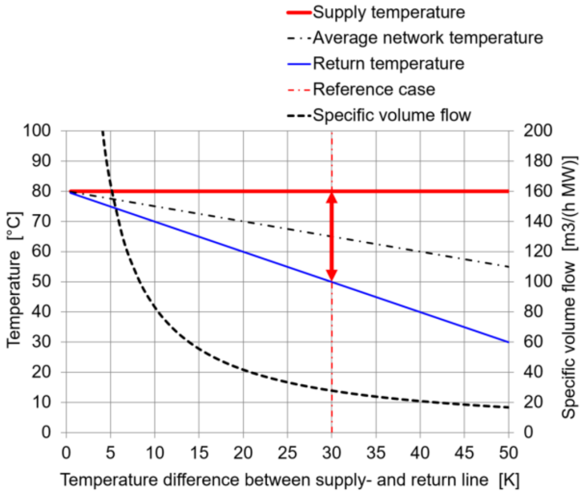


Figure 1.2 Specific volume flow as a function of the temperature difference between flow and return (scale on the right) . The specific volume flow corresponds to the volume flow for a district heating network with 1 MW at the network input. Supply and return temperatures for a supply temperature of 80 °C are shown and as an example is network with a temperature difference of 30 K

4. The **temperature level** of the grid affects efficiency and costs in several ways. By raising the temperature level with constant temperature difference, the heat losses of the network increase. However, heat generation efficiency can decrease. This indirect effect on heat generation can be much greater than the direct effect on heat losses and is particularly pronounced for the following applications:
   a) For heat pumps as well as for plants for combined heat and power generation with steam processes (steam turbines or ORC plants). In these cases, the supply temperature is the most important factor, since the coefficient of performance of the heat pump or the electricity yield of the cogeneration system decreases when operating tem-perature is increased.
   b) For boilers with exhaust gas condensation. In this case, the return temperature is decisive, since efficient heat recovery through flue gas condensation only occurs when the temperature falls significantly below the dew point of the flue gas.
5. As the **heat transfer stations** and the **heat consumption** influence the operation and efficiency through the return temperature, the type and kind of heat transfer on the consumer side (i.e. the house substations) must also be taken into account for the cost-effective operation of a district heating network. Possible measures to avoid nega-tive effects on the network and heat generation are to include requirements for a minimum temperature difference or a maximum return temperature in the heat supply contract. Furthermore, the monitoring of the grid tempera-tures enables measures to be taken in the event of deviation from the design specifications.

The economic efficiency of a district heating network is thus determined by a variety of factors. An analysis of the influence of the design parameters on the economic efficiency shows that in order to optimise the heat distribution costs at given temperatures and connected loads, it is crucial that the pipelines are designed for the **smallest technically permissible pipe diameter** [14]. Figure 1.3 shows the heat distribution costs for an example with the following assumptions:

- 1 MW connected load
- 1 km pipeline length
- Network operation all year round
- Heat consumer 2'000 full-load operating hours p.a.
- Linear heat density at the system input is therefore 2.0 MWh/(Trm a). With 10% grid losses, this corresponds to 1.8 MWh/(Trm a) at the grid output.
- Supply temperature 80 °C
- Temperature difference 30 K
- Annuity 5.1 % p.a. (30 years, 3 % p.a.)
- Heating costs at network input 60 CHF/MWh
- Maximum flow velocity according to ÖKL Merkblatt 67 (Figure 1.4), [120]), which corresponds to a pressure loss of about 200 Pa/m in the determined area.

The heat distribution costs are described as a function of the nominal diameter and have a minimum that coin-cides with the smallest permissible nominal diameter. The minimum heat distribution costs are equal to 26

CHF/MWh of heat supplied to the heat consumer. A design for a nominal diameter larger than necessary increas-es the cost of heat distribution by 9 %, two nominal diameters larger by 30 %.

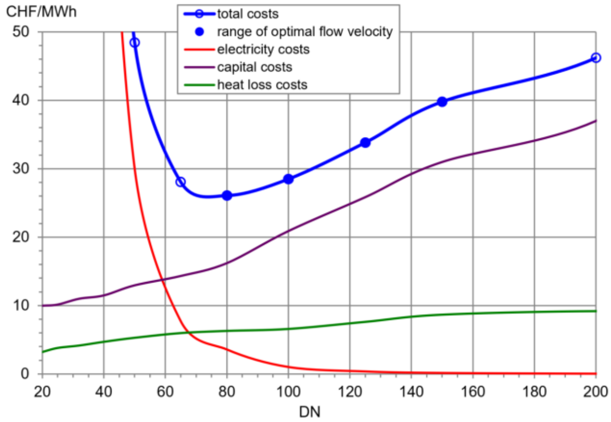


Figure 1.3 Heat distribution costs as total costs and divided into capital, heat loss and electricity costs as a function of the nominal diameter [14]. The filled in points show the smallest permissible nominal diameter and the next three larger ones.

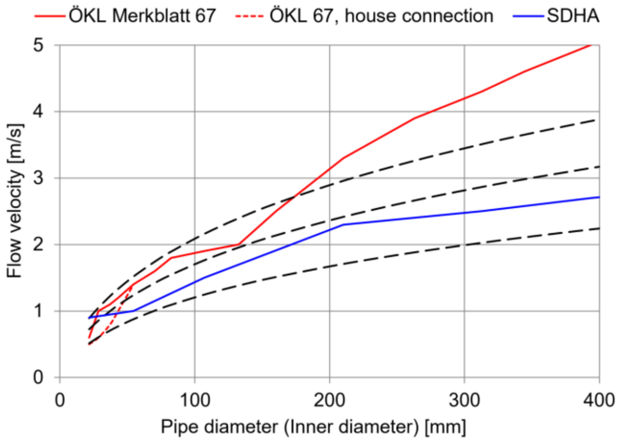


Figure 1.4 Flow velocities as a function of the pipe diameter (inside diameter): Recommendations for maximum flow velocity according to ÖKL leaflet 67 [120] and recommendations of the Swedish District Heating Association (SDHA) [3]. The calculated flow velocity corresponding to pressure losses of 100, 200 and 300 Pa/m are shown [14].

The smallest permissible nominal diameter is determined by the maximum permissible flow velocity, which ensures operation without unacceptable cavitation and noise generation. Figure 1.4 shows the corresponding recommendations according to ÖKL leaflet 67 [120] and the Swedish District Heating Association (SDHA) [3]. In addi-tion, the flow velocities corresponding to a pressure drop of 100, 200 and 300 Pa/m are shown [14]. A practical survey of 52 district heating networks shows that around 80% of the main and sub-tracks are designed larger than actually required [16]. The oversizing usually corresponds to one or two, but occasionally up to four nominal di-ameters, and causes significantly higher heat losses and costs compared to a network with the smallest possible pipe diameter.

Other important factors are the temperature difference and the temperature level. An increase in the **temperature difference** reduces heat distribution losses for the same pipe diameter, as Figure 1.5 shows for the example described. Figure 1.5 also shows that an increase in temperature difference allows the use of smaller pipe diameters. This leads to a further reduction in heat losses and at the same time results in significantly lower investment costs. For this reason, the total heat distribution costs can be significantly reduced by increasing the temperature difference, as shown in Figure 1.7. Finally, a large temperature difference at a given supply temperature leads to low return temperatures, which additionally increases the efficiency of heat generation. In existing plants, there is considerable potential for improvement in terms of temperature difference, as a practical survey has shown that many networks also have higher return temperatures [16]. This leads to increased auxiliary energy consumption and can also increase the energy consumption of the heat generator.

High supply temperatures increase heat losses and thus also operating costs (Figure 1.6). While the dimensioning must be based on the minimum and thus optimum pipe diameter, it must be ensured that the network temperatures are maintained. If the connected load is reduced by a too small temperature difference, optimisation on the side of the heat consumers may increase the connected load. Correct regulation of the house substations may improve performance, otherwise an enlargement of the heat exchangers is necessary.

Another parameter that influences heat losses and costs is the **thermal insulation**. In contrast to the pipe diameter and the temperature difference, the thermal insulation has opposite effects, reducing operating costs but increasing investments. As Figure 1.8 shows, there is a clear difference in losses, especially between thermal insulation class 1 (minimum) and class 2 (medium). Thermal insulation class 1 is not suitable for district heating networks within Switzerland. Thermal insulation class 2 produces, for the described example with the minimum diameter, losses of around 10.5 %. A target value based on the “QM- Holzheizwerke” of 10 % is reached by thermal insulation class 3 [21]. The minimum possible diameter or one more nominal width class should be used.

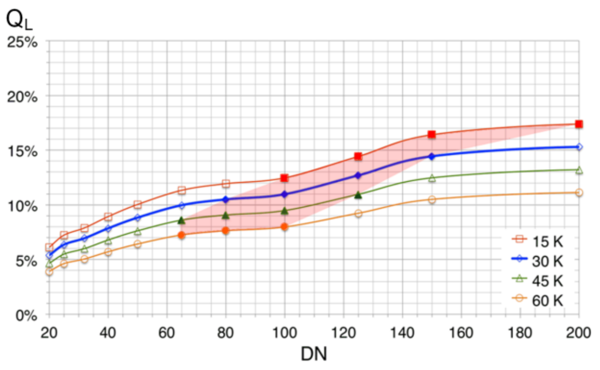


Figure 1.5 Heat distribution losses as a function of the nominal diameter for varying temperature differences in the district heating network described in Figure 1.3 [14].
Highlighted area: nominal diameter with permissible flow velocity.

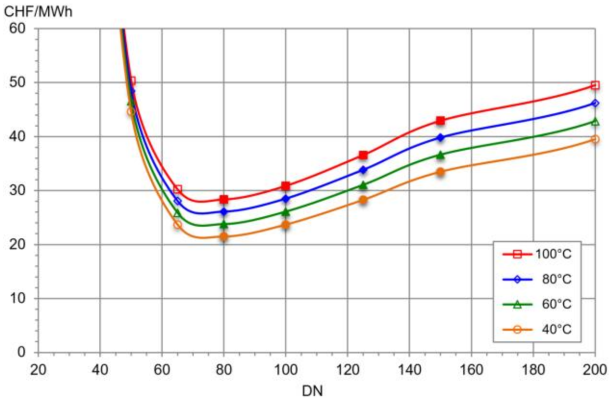


Figure 1.6 Heat distribution costs as a function of the nominal diameter for varying supply temperatures with a temperature difference of (30 K) [14].

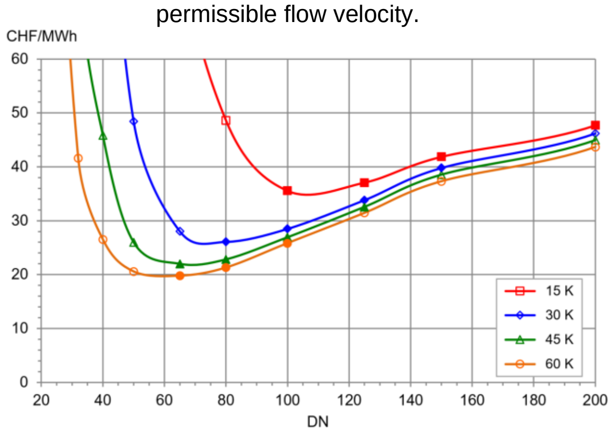


Figure 1.7 Heat distribution costs as a function of the nominal diameter for varying temperature differences at a supply temperature of 80 °C for the district heating network described in Figure 1.3 [14].

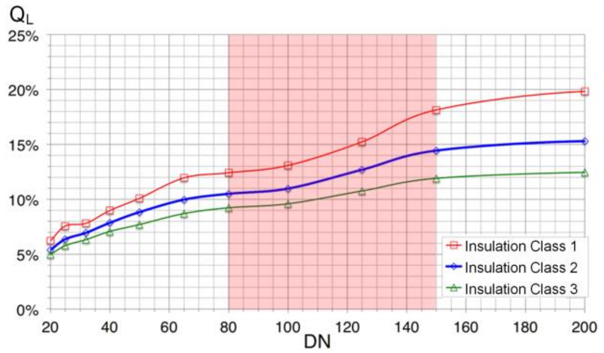


Figure 1.8 Heat distribution losses as a function of the nominal diameter for various thermal insulation classes. Highlighted area: The nominal diameter with permissible flow velocity [14].

The most important influences on the cost efficiency of district heating networks can thus be summarised as follows:

1. The investment costs, network losses and auxiliary energy consumption of the district heating network are determined by the following parameters:
   - **pipe diameters** (and thus the dimensioning of the network)
   - **pipe type** (material, design)
   - thermal **insulation class**.
2. The efficiency of heat production is influenced by the following parameters of the district heating network:
   - **temperature difference**
   - **temperature level** of the supply and return temperature.

2 Heat supply

This chapter deals with the basic features of heat demand and heat load demand and describes the division of the annual duration curve into base load and peak load. In addition, the most important heat generators are outlined. Their influence through operation and the integration of heat storage tanks is explained. For heat demand, a distinction is made between process heat and the heat supply of buildings. The heat demand of residential buildings includes space heating and hot water supply.

2.1 Heat demand of individual consumers

2.1.1 Space heating

The **annual heat requirement for space heating** in new buildings is calculated on the basis of “SIA-Norm 380/1” [81]. This takes into account heat gained internally from solar radiation, people, electrical equipment and other sources. For existing buildings, the calculation is usually based on the previous energy consumption, for example the annual oil consumption and the annual efficiency of the existing heat generator, although this must be divided into space heating, hot water and processes. If reliable data on past consumption or a breakdown into space heating, hot water and processes is not possible, measurements can be taken or estimated.

The standard **heat load demand for space heating** is calculated for new buildings according to SIA 384.201 [82]. In contrast to SIA 380/1, no heat gains are taken into account, as their occurrence over time cannot be controlled or predicted. For existing buildings, the best method for determining the heat load demand is to determine the load characteristics on the basis of measurements, which is particularly useful for large consumer loads and process heat [21]. However, measurements are only possible if sufficient time and a functioning heat generation system are available. Reliable calculations of heat load demand are rarely available or the data is based on outdated calculation methods. For new calculations, however, detailed plans and information on wall construction and other parameters are often missing. The most common method is therefore to estimate the heat load demand from the previous heat consumption. The maximum heat load demand for space heating is obtained by dividing the heat demand by a suitable number of full operating hours. The number of full operating hours depends on climate, the design limit, the use of the building (residential or other) and the non-weather dependent part.

Upcoming energy saving measures can influence the annual heat demand, the heat load demand and the temperature demand and have to be considered accordingly in the design. In the case of buildings, it is crucial that the proportion of space heating is significantly reduced through better building technology, while the demand for hot water is hardly affected. For example, in the years around 1970, hot water heating accounted for only just under 10% of the heat demand, while values of up to around 50% can be expected in the future, as explained in chapter 2.4. The renovation of existing buildings can therefore lead to a reduction in the space heating demand during the operating time of a district heating network and, together with the connection of new buildings, cause a significant reduction in the proportion of space heating.

2.1.2 Domestic hot water and process heat

The calculation of the annual heat demand for **domestic hot** water in new buildings is usually based on standard use. For existing buildings, the previous energy consumption and an estimate or measurement of the proportion for hot water can be used.

The average value for the heat load demand for hot water in new and existing buildings is calculated by dividing the hot water heat demand by the annual number of heating hours in the case of winter operation or by 8760 annual hours in the case of year-round operation. The peak value of the heat load demand for hot water is calculated from the connected load of the water heater. Storage tanks are usually used for domestic hot water in Switzerland, while instantaneous water heaters are rare, and therefore the annual hot water demand is usually divided by 4000 to 6000 full operating hours for residential buildings. This takes into account a peak load which is about twice as high as the average value at 8760 annual hours with constant output. This is necessary because the peak load for the hot water demand can be higher on certain days, as the hot water demand can depend on the day of the week and the season, for example.

The average heat load demand for **process heat** is calculated by dividing the heat demand by the annual operating hours of the process heat consumer, which is usually determined by means of an operating hours meter. A higher peak load is also taken into account for process heat if required.

2.1.3 Temperature requirements

The temperature demand, i.e. the minimum required supply temperature, depends on the heat transfer design, the type of domestic hot water preparation and other factors. The design of radiators or underfloor heating and other heat exchangers is usually based on the manufacturer's specifications and is known for new buildings. In existing buildings, it is possible to estimate the temperature demand on the basis of the existing heat consumers (use of radiator or underfloor heating, water heaters, etc.). However, it is recommended to carry out temperature measurements at the heat consumer at low outside temperatures and to extrapolate the measured value pairs (flow/return temperature, outside temperature) to design specifications.

2.2 Total heat demand

When determining the heat load demand of the entire system from the data of the individual heat consumers, the following challenges may arise which must be taken into account in the design:

1. The heat load demand for the entire system results from a combination of calculated values with more or less large safety margins and real measured values without safety margins.
2. The standard heat load demand for space heating according to SIA 384.201 is based on a standard ambient temperature and does not take into account internal heat gained [82], whereas load curves determined by measurements are based on real outside temperatures and include heat gained.
3. The number of full operating hours is required to estimate the heat load demand for space heating from the heat demand of existing buildings. Since this number depends on various factors such as the annual duration curve of the outside temperature at the location of the system, the room temperature, the heating limit and the size of the non weather-dependent component, its estimation is uncertain.
4. Heating capacities to compensate for intermittent heating (e.g. heating of office buildings on Monday morning after reduced weekend operation) are usually not taken into account.
5. Metrologically determined load characteristics can be created for different load cases by regression of daily averages up to 1-hour averages. However, it must be noted that measured peak loads are not only dependent on the heat consumer, but are also influenced by limitations of the heat generator or by the system inertia.
6. Existing buildings often have a considerable proportion of room heat demand that is not weather-dependent, which can amount to 5 % to 10 %, for example due to poorly insulated distribution, and which appears in the load characteristic curve as a power jump at the heating limit. While it is possible that measured load curves of old buildings show a large weather-dependent share of room heat demand, this is almost non-existant in new buildings.
7. As explained in Chapter 2.1, a correct dimensioning of the heat load demand for hot water must be carried out, as the peak load can be significantly higher than the average value determined over the year.

In order to determine the figures for the entire system as realistically as possible from a combination of calculations and real measured values, the following questions must therefore be clarified:

1. How is gained heat in new buildings taken into account?
2. What are the appropriate full operating hour figures for determining the heat load demand for space heating based on the previous heat demand?
3. How can the proportion of the heat load demand for space heating that is not weather-dependent be taken into account?

2.3 Heat load demand

The annual load duration curve of the heat load demand is used for an approximate design of the heat generation. This is based on

1. the load characteristic curve of the overall system according to Figure 2.1
2. the annual load duration curve of the outside temperature according to Figure 2.2.

The information from these two graphs is combined in the annual load duration curve of the heat load demand shown in Figure 2.3.

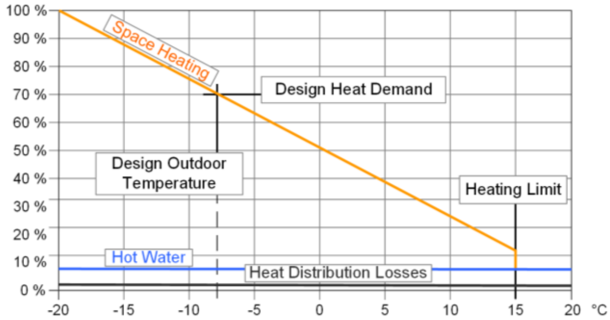


Figure 2.1 Stacked load characteristic curve of the heat load demand of the entire system as a function of the outside temperature [21].

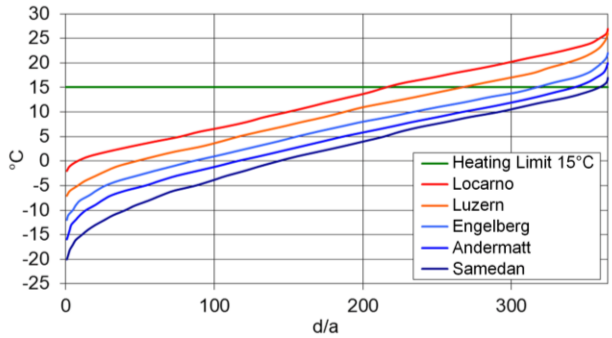


Figure 2.2 Outdoor temperature represented as a 10-year average daily value from 2002 to 2011 for different locations in Switzerland [21]. A heating limit of 15 °C is shown in green.

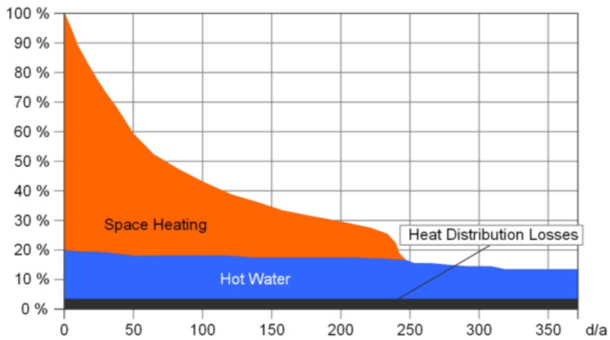


Figure 2.3 Characteristic annual load duration curve of the heat demand for a district heating network.

The representation of the heat load demand as a load characteristic curve (Figure 2.1) with outside temperatures that are as real as possible comes from practical experience with measurements taken during the renovation and expansion of larger building services systems. It requires empirically supported decisions such as the choice of the design heating limit. The advantage of this representation is that data from calculations from previous energy consumption and from measurements can be clearly combined in one graph.

The load characteristic curve of the entire system is obtained by stacking the individual load characteristic curves for room heat, hot water, process heat and heat distribution losses. The heat load demand of the entire plant at a specified temperature can be read from it.

The annual load duration curve of the outdoor temperature is the representation of the cumulative frequency of the outdoor temperature as the number of days per year. From Figure 2.2, for example, it can be deduced that the 10-year daily mean value of the outside temperature in Lucerne was below 4 °C on 100 days. For the outside temperature, the 24-hour average value is always to be used, whereas the heat load demand can be a daily average value (e.g. for residential buildings) or a peak value (e.g. for a bank building).

The annual load duration curve of the heat load demand for the entire system is also obtained by stacking several annual load duration curves for space heating, hot water, process heat and heat distribution losses. Figure 2.3 shows an example of the annual load duration curves for space heating, hot water and heat distribution losses.

In the example, the heat losses of the network are assumed to be constant over a year for simplification purposes, as the temperature difference between the flow and return flow and the ground temperature around the district heating pipe change only slightly over the course of a year.

The second component describes the heat demand for hot water throughout the year, whereby in the example a slightly reduced demand in summer is seen. For the sake of simplicity, however, the hot water demand is usually assumed to be constant throughout the year.

The third component describes the space heating demand as the largest share of the total heating demand.

2.4 Demand for old and new buildings

Space heating and domestic hot water in buildings make up an important part of the heat consumption for district heating networks. As this is influenced by the building standard, the annual load duration curves for residential buildings are shown below. These describe a typical building standard from 1970 compared to a modern one. The modern standard is represented as the year 2020 and corresponds approximately to a building according to the Minergie® standard of 2015 (low-energy building) or according to the model regulations of the cantons in the energy sector (MuKEn), whose introduction is planned for 2020 [71]. The comparison is based on the values in Table 2.1. The heat output ratio in W/m² and the energy ratio in kWh/(m² a) are shown. The resulting number of full operating hours for space heating and hot water can be seen, as well as the overall figures. The building from 1970 has a heat load demand of 85 W/m² under specified conditions (-7 °C, Zurich), compared with 25 W/m² for the building from 2020.

The representation of the annual load duration curves is based on the average daily temperatures for Zurich based on the specifications described in “QM Holzheizwerke” [21]. For a building of the year 1970 this results in annual load duration curves of the heat load demand as shown in Figure 2.4. The graphic on top represents the absolute value of the heat load demand in W/m². As example is presented a building with 1000 m² energy reference area and the value reflects just the heat load demand in kW. The graphic at the bottom shows that the heat load demand is standardized to 100 % in order to represent the progression of the represented building on the right side. Figure 2.5 represents the behavior of a building by standards of the year 2020.

Table 2.1 Performance and energy indicators of residential buildings (example) with 1970 and 2020 standard for space heating (SH) and domestic hot water (DHW) per m² energy reference area and designed for -7 °C (Zurich).

	Unit	1970	2020
Space heating demand (SH)	W/m²	80	20
	kWh/m²a	185	20
Full -load operating hours SH	h/a	2300	1000
Domestic hot water demand (DHW)	W/m²	5	5
	kWh/m²a	20	20
Full-load operating hours DHW	h/a	4000	4000
Demand SH+DHW (= 100 %)	W/m²	85	25
	kWh/m²a	205	40
Full-load operating hours SH+DHW	h/a	2400	1600
Heating limit	°C	15	10

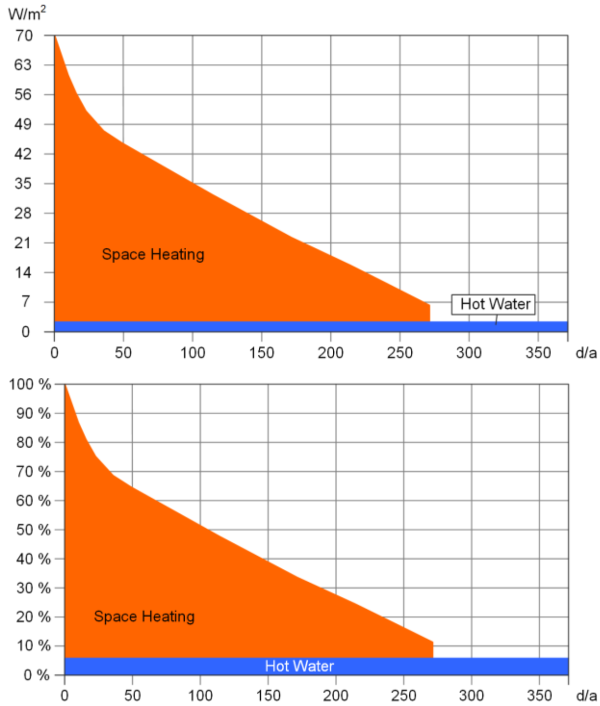


Figure 2.4 Annual load duration curve of the heat load demand for a residential building from 1970 for domestic hot water and space heating (design according to [21] for Zurich, daily mean temperature -7 °C).
Above heat load demand in [W/m^2].
Below in percent with 100 % = maximum value

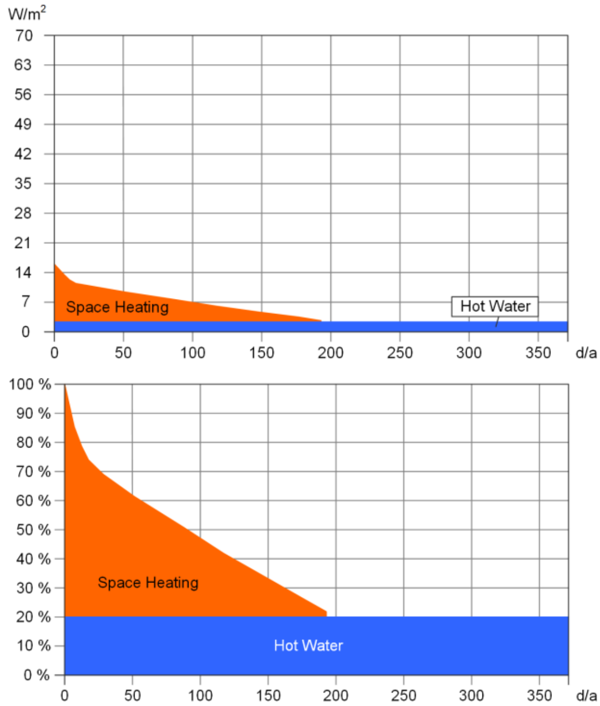


Figure 2.5 Annual load duration curve of the heat load demand for a residential building in 2020 for domestic hot water and space heating (design according to [21] for Zurich, daily mean temperature -7 °C).
Above heat load demand in [W/m^2].
Below in percent with 100 % = maximum value

The comparison of the graphs shows that the heating limit is significantly reduced by the improved building standard and the heating period is about 50 days shorter. According to Table 2.1, the energy demand for space heating and hot water drops from 205 kWh/(m^2 a) to 40 kWh/(m^2 a) or to 20% of the initial value. In contrast, the share of hot water increases from 6% in old buildings to 20% in new buildings. The share of energy consumption for hot water increases from just under 10% to 50%, which is also forecast by other studies [43].

The fact that a new building has a significantly higher base load share for hot water than an old building is advantageous for the summer operation of a district heating network. However, as the number of full operating hours for space heating in new buildings falls to less than half of that of old buildings, the new building still has a lower average load than an old building (regarding space heating and hot water, 1600 instead of 2400 full operating hours per year), which is disadvantageous for a district heating network.

Therefore, when designing district heating networks, it must be taken into account that the building structure has a significant influence on the heat demand and that this can change during the operating time due to building renovations. If heat sales decrease, the percentage of heat distribution losses and capital costs increase, while the connection density decreases. For supply areas with low-energy buildings, the low connection rates are disadvantageous for district heating, whereas the hot water and thus the base load share is significantly higher in summer.

2.5 Design specification heat generator

When designing the heat generation system, it must be decided whether a single or several heat generators are used to cover the heat demand and whether the output is distributed to one or several energy carriers.

A heat generator must be designed for the maximum heat load demand. To supply buildings, this means the heat generator can cover a correspondingly large load range either directly or by integrating a heat storage tank. For the hot water supply in summer, the minimum output for old buildings is only around 6%, while the figure for new buildings is 20% For the district heating network in both cases the heat distribution losses must also be covered. The resulting output range thus often covers a factor of 5 or more, depending on the consumer and the network. To cover this performance range, the use of two or more heat generators is generally advantageous. In the case of automatic wood-fired heating systems, this can result in better plant operation with lower emissions and lower heat losses from the boilers, as well as cost reduction for auxiliary energy and maintenance.

The basis for the design is the stacked annual load duration curve of the heat demand, which is determined by calculation, measurement or comparison with similar district heating networks, as described in Chapter 2.3. The area below the annual load duration line represents the annual heat demand, which can be divided into base load and peak load as shown in Figure 2.6. The base load is characterized by a high number of operating hours, while the peak load describes the extra heat load demand with a low number of operating hours.

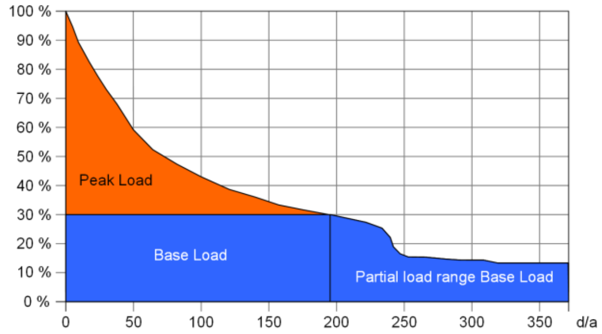


Figure 2.6 Breakdown of the annual duration curve into base and peak load.

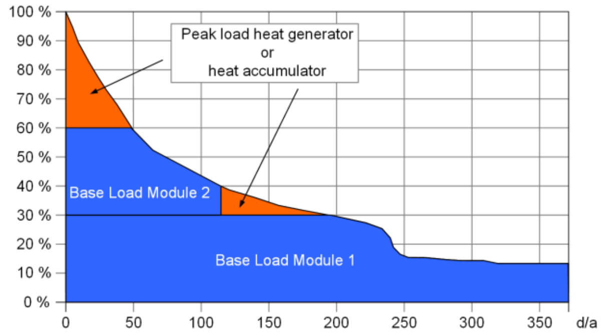


Figure 2.7 Covering the base load with several base load modules.

Depending on the heat generator, the economic performance share of the base load usually accounts for between 10 % and 60 % of the maximum performance, which usually covers between 50 % and 90 % of the total annual heat demand [43]. Figure 2.6 shows a share of 30 %, which can correspond to a biogas-operated combined heat and power unit (CHP), for example, where summer operation is carried out with reduced load or (where permissible) with only partial heat utilisation.

If the base load heat generator(s) can only be operated to a limited extent for technical or economic reasons, the peak load heat generator may cover the low load, as shown in the example in Figure 2.7. This can be useful, for example, for an automatic wood-fired boiler, since low-load operation can lead to increased emissions, especially when wet fuels are used.

For larger units, coverage of the base load energy carrier can also be increased by multi-modular base load systems, thus achieving additional redundancy (Figure 2.7).

As a further measure, the integration of large heat storage units is being used more frequently. This reduces peaks in the heat load demand and thus increases the base load share. In the case of combined heat and power generation, heat storage also enables temporary current-controlled operation, which increases the electric input and reduces peaks in electricity consumption [43].

If a waste heat source is available for heat supply, it can be used as a cost-efficient solution to cover the base load in advance. This also applies to waste heat from cogeneration plants, which are largely heat-guided wherever possible. If no waste heat is available, technologies with high capital costs and low operating costs are often used for the base load. This is particularly true for automatic wood-fired systems, as well as solar thermal systems which have high capital costs and relatively low fuel costs. The integration of solar heat is therefore usually limited to the base load and can only cover part of the additional heating demands.

Up to now, boilers with natural gas or heating oil have often been used to cover peak loads and for possible recovery. In order for district heating to have a clear advantage over decentralised oil and gas heating in terms of CO2 emissions, a fossil peak load share must be minimised. As a rule, a share of 80 % to 90 % of the annual heat demand can be met by base load heat generators.

In summary, the conditions in Table 2.2 are typical for the combination of base load and peak load heat generators.

Table 2.2 Typical characteristics for base load and peak load heat generators.

Base Load	Peak Load
Operation by nominal- or partial load for a limited load range	Common operation in partial load range
Extended degree of utilization respectively high amount of full-load operating hours	Low capacity utilization or small number of full-load operating hours
Few and slow load changes	Often and fast load changes
Few start-up and shut-down operations	Frequent start-up and shut-down operations

2.6 Evaluation of heat and electricity

2.6.1 Efficiency rate and degree of utilization

The **efficiency rate** of a plant is defined as the ratio between usable energy and energy supplied. Under stationary or pseudo-stationary conditions without distortions by energy storage, it can also be determined as the ratio between the momentary usable power and the supplied power in kW/kW at a given moment.

In this handbook, the term efficiency rate is used for a reading of performance at a given moment or for a value determined over a short period of time. Examples are the mechanical efficiency of a motor at one load point or the boiler efficiency of a continuously operating boiler at a steady state of constant performance. The efficiency determined in this way is valid either for one load point or for a specific operating cycle. An example of an operating cycle is the combustion of a manually fed wood log boiler, which takes several hours. During the combustion process, the operation is unsteady, but distortions due to storage effects are compensated by returning to the initial thermal state. In principle, a combustion engine is also considered over at least one cycle, i.e. over four cycles in a four-stroke engine. However, the cycle duration is much shorter for an engine, which is why operation appears pseudo-stationary even at a time resolution of one second.

For an assessment over a longer period of time, the term **degree of utilisation** is used. The rate of capacity utilization is defined as the ratio between the total useful performance over a longer period of time and the total input energy over a given period of time. A year is often chosen as the analysis period and the value determined is then referred to as the **annual utilisation rate**. The annual utilisation rate is therefore the ratio of the energy produced in one year to the energy supplied in one year in (kWh/a)/(kWh/a).

Heat pumps also differentiate between momentary values and the performance added up over a longer period. For example, the performance factor of a heat pump describes the ratio between useful heat and supplied electrical power as an momentary value, while the annual performance factor (APF) describes the ratio between annual heat production and annual electricity consumption.

Efficiency data may describe a single unit such as an engine or boiler or entire plants or systems. For better assessment, limits must therefore be defined precisely. In the case of electric power plants, for example, the own electricity consumption must be taken into consideration. Thus, the electricity production at the generator corresponds to the gross production, whereas after deduction of the own electricity consumption, the net production results. Gross and net values can be identified for both efficiency and degree of utilization.

2.6.2 Overall efficiency

The overall efficiency is the sum of electric efficiency and heat utilisation efficiency. In connection with district heating, thermal plants with steam turbines or ORC plants are mainly used to generate electricity. Such steam processes achieve maximum electrical efficiencies if the condensation of the process medium takes place at the lowest possible temperature. For this purpose, the waste heat can be dissipated to the environment, for example, via a cooling tower. If, on the other hand, the waste heat is to be used, the condensation must take place at a higher temperature, which reduces the electric efficiency. To differentiate between the applications, efficiencies of heat and power generation are shown in Figure 2.8.

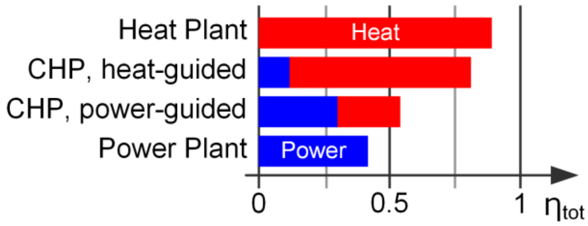


Figure 2.8 Overall efficiency with shares of heat (red) and electricity production (blue) for heating plant, combined heat and power plant and power plant. The diagram shows the qualitative behaviour and corresponds, for example, to the following types of plant (from top to bottom): 0.5–10 MW_q Wood-fired Heating Plant, 0.5 MW_{el} Wood-fired Co-generation Pl., 20 MW_{el} Wood-fired Co-generation Plant, 500 MW_{el} Coal-fired Power Plant.

There is no waste heat recovery in a pure power plant. The electric efficiency is maximal and corresponds to the overall efficiency. Because of the losses for the conversion of heat into electricity, the overall efficiency is lower than in plants with heat utilisation.

A CHP plant is a thermal plant with combined heat and power generation. For steam plants, the electric efficiency is reduced because of the heat needed for steam. If the cogeneration plant is operated with electricity, part of the heat is used, which leads to a reduction in electric efficiency, while the rest is released into the environment. In heat-operated CHPs, all of the waste heat is used, further reducing the electric efficiency. At the same time, the overall efficiency increases with increasing heat utilisation.

A heating plant produces only heat and achieves the highest overall efficiency.

For momentary values or short-term observations, the following conditions apply to the efficiency of a co-generation plant:

$$\eta_q = \frac{\dot{Q}_K}{\dot{Q}_{fuel}}$$

$$\eta_{el} = \frac{P}{\dot{Q}_{fuel}}$$

$$\eta_{tot} = \frac{\dot{Q}_K + P}{\dot{Q}_{fuel}} = \eta_q + \eta_{el}$$

η_q = Efficiency Rate Heat Production [–]
η_{el} = Efficiency Rate Electricity Production [–]
η_{tot} = Overall Efficiency [–]
P = Electricity Production [kW]
$\dot{Q}_K$ = Usable Heat from Boiler/Kettle Output [kW]
$\dot{Q}_{fuel}$ = supplied Heat Load in Fuel[kW]

For a power plant:

$$\eta_q = 0$$

$$\eta_{tot} = \eta_{el}$$

For a heating plant:

$$\eta_{el} = 0$$

$$\eta_{tot} = \eta_q$$

For a boiler, the efficiency of heat production becomes the boiler efficiency :

$$\eta_q = \eta_K$$

η_K = Efficiency of the Boiler [–]

For a steam boiler used to drive a steam process a part of the boiler capacity is used for electric power generation by $\eta_{el} > 0$. In this case:

$$\eta_q < \eta_K$$

For evaluations over longer periods of observation, the conditions apply analogously to the **degree of utilization**. These are described using the example of the **annual efficiency** and marked with the index a:

$$\eta_{q,a} = \frac{Q_{K,a}}{Q_{fuel,a}}$$

$$\eta_{el,a} = \frac{E_{el,a}}{Q_{fuel,a}}$$

$$\eta_{tot,a} = \frac{Q_{K,a} + E_{el,a}}{Q_{fuel,a}} = \eta_{q,a} + \eta_{el,a}$$

$\eta_{q,a}$ = Annual efficiency heat generation [–]
$\eta_{el,a}$ = Annual efficiency electricity production [–]
$\eta_{tot,a}$ = Overall annual efficiency [–]
$Q_{K,a}$ = Annual heat production [kWh/a]
$E_{el,a}$ = Annual electricity production [kWh/a]
$Q_{fuel,a}$ = Annual supplied fuel energy [kWh/a]

For a central heating boiler, the annual efficiency of heat production becomes the annual efficiency of the boiler:

$$\eta_{q,a} = \eta_{K,a}$$

$\eta_{K,a}$ = Annual efficiency of the boiler [–]

In order to achieve a high overall utilisation of the fuel, two types of applications are possible:

- Either high electric efficiencies are aimed for and operation without or only partial heat utilisation is permitted. Such plants can be electrically operated and contribute to the power supply even if the heat demand decreases. One example of this is natural gas-fired combined cycle power plants, which achieve an electric efficiency of around 60 % at outputs of over 500 MW_{el}.
- Plants with low electric efficiencies but with total waste heat utilisation can be used. Such plants are largely heat-operated and are therefore designed for base load heat demands in combined heat and power plants. In Switzerland, such plants are mainly constructed for an output of less than 10 MW_{el}. Examples are wood-fired CHP plants with steam turbines or ORC modules.

2.6.3 Weighted overall efficiency

As electricity is more valuable than heat, a weighting factor for electricity is introduced for the comparison of heat and electricity production, thus defining a **weighted overall efficiency** for a CHP plant as follows:

$$\eta_{tot,weighted} = \frac{\dot{Q}_K + f_{el} \cdot P}{\dot{Q}_{fuel}} = \eta_q + f_{el} \cdot \eta_{el}$$

$\eta_{tot,\ weighted}$ = Weighted overall efficiency [–]
f_{el} = Weighting factor for electricity [–]

From this formula, the weighted overall efficiency can be derived. This is described by index a, which corresponds to the weighted total utilisation ratio over one year. However, the explanations also apply to other observation periods, which is why the term "**weighted total utilisation factor**" is used in the following to simplify matters. The following therefore applies:

$$\eta_{tot,a,weighted} = \frac{Q_{K,a} + f_{el} \cdot E_{el,a}}{Q_{fuel,a}} = \eta_{q,a} + f_{el} \cdot \eta_{el,a}$$

$\eta_{tot,a,weighted}$ = Weighted total utilization factor [–]

For $f_{el} = 1$ is valid for: $\eta_{tot,ev} = \eta_{tot}$ and $\eta_{tot,a,ev} = \eta_{tot,a}$.

An effective weighting of the electricity is achieved if a value of $f_{el} > 1$ is used. The weighted overall efficiency can then also reach values > 1 and must not be compared with unweighted values.

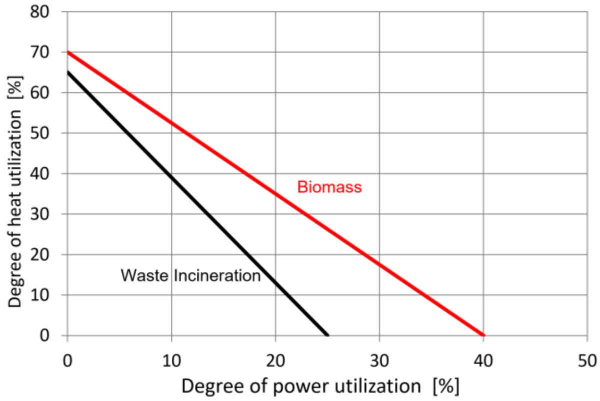


Figure 2.9 Minimum requirement of the Energy Regulation for Steam Processes ("ENV Annex 1.5, Paragraph 3.3 (KVA) and Paragraph 6.3 Letter a (Biomasse)" [75]) for the remuneration of cost-covering feed-in tariffs for electricity from biomass (Figure 2.9, Article 7a EnG, and "Biomasse Anh. 1.5 EnV" [76]). The given period is one year and the gross electricity production can be evaluated.

Electricity is also effectively weighted, for example, in the minimum requirements for cost-covering feed-in tariff (CCFT) from thermal plants [75]. Figure 2.9 shows the minimum CCFT requirements for steam processes fired with biomass or installed in waste incineration plants [76] as required by the Energy Regulations (ENV) [76]. The requirements for biomass plants are much higher than for waste incineration plants (WIP).

In [76] it is stated that the utilisation rates are to be assessed over a period of one year and that the total electricity production may be assessed without deduction of own consumption. The CCFT requirements thus correspond to a weighted overall efficiency of gross electricity production over one year.

For **Biomass** the requirements of the energy ordinance in Switzerland (ENV) are:

$$\eta_{tot,a,weighted} = \left(\eta_{q,a} + f_{el} \cdot \eta_{el,a}\right) \geq 70\%$$

With f_{el} = 70 % / 40 % = 1.75.

For **WIP** these are:

$$\eta_{tot,a,weighted} = \left(\eta_{q,a} + f_{el} \cdot \eta_{el,a}\right) \geq 65\%$$

With f_{el} = 65 % / 25 % = 2.60.

The ENV thus introduces a weighting of electricity with a factor of 1.75 or 2.60.

Figure 2.10 shows the influence of the electric efficiency and the weighting factor on the weighted overall efficiency for a heating plant, a combined heat and power plant and a power plant. For the weighting factor, a base value of 1.75 (ENV) for biomass is assumed as well as a value of 3.5, which is twice as high. The values with efficiencies achievable in real plants are shown in bold and cover electric efficiencies of 10 % to 40 % for a CHP plant and 30 % to 60 % for a power plant. Heat-guided operation is assumed for the co-generation plant.

The graph shows that when electricity is weighted with a factor of 3.5, both a CHP plant and a power plant achieve a higher weighted overall efficiency than a heating plant. With a weighting of 1.75, heating plants and CHP plants achieve similar overall values, whereas a power plant only achieves equally high or higher values from an electric efficiency of over 50 %.

The electricity weighting factor introduced for electricity production can also be used to evaluate consumers and can be compared with the coefficient of performance (COP) or annual performance factor (APF) of a heat pump or interpreted as such. For applications other than a heat pump, such as the electricity consumption of a heat recovery system, the weighting factor can also be interpreted in general as a factor of electro-thermal enhancement (ETE).

The **Performance Factor** ε or the Coefficient of Performance (COP) of a heat pump relates to:

$$\varepsilon = \frac{\dot{Q}_{UH}}{P_{el}} = \frac{\dot{Q}_{sply} + P_{el}}{P_{el}} = 1 + \frac{\dot{Q}_{sply}}{P_{el}} = \frac{T_{up} - T_{low}}{T_{up}} = 1 - \frac{T_{low}}{T_{up}}$$

$\dot{Q}_{UH}$ = useful heat capacity [kW]

$\dot{Q}_{sply}$ = supplied heat capacity [kW]

P_{el} = supplied electrical power [kW]

T = upper temperature (Utilization Heat) [K]

T_{low} = lower temperature (supplied Heat) [K]

The index «low» denotes the heat input by ambient or waste heat at the lower temperature level Tlow.

The coefficient of performance theoretically achievable between two temperature levels is called Carnot coefficient of performance ε_C, because the process describes a counterclockwise cycle of the Carnot process. Figure 2.16 shows the influence of temperatures on the Carnot coefficient.

The ratio between the real and theoretical coefficient of performance is called the quality grade η_{QG} and reaches typical values of $\eta_{QG} = \varepsilon/\varepsilon_C$ = 0.4 – 0.55.

The following applies to the **annual performance factor (APF)**:

$$APF = \frac{Q_{UH,a}}{E_{el,a}} = \frac{Q_{sply,a} + E_{el,a}}{E_{el,a}}$$

$Q_{UH,a}$ = Annual produced useful heat [kWh/a]

$Q_{SPLY,a}$ = Annual supplied heat [kWh/a]

$E_{el,a}$ = Annual Electricity Consumption [kWh/a]

As a comparison to a reference case with decentralised and fuel-fired heating systems, a scenario can be considered in which a CHP plant is operated with the fuels previously used in heating systems. The heat is used for district heating, for example, while the electricity is used to drive decentralised heat pumps that use ambient heat as a heat source and raise it to the usable temperature. The energy flow for this scenario is shown in a Sankey diagram in Figure 2.11. For the cogeneration plant, a degree of utilisation of 60% for heat and 20% for electricity is assumed, which corresponds to an overall degree of utilisation of 80%. If an annual performance factor of 3.5 is assumed for heat pumps and this is interpreted as a weighting factor for electricity, a weighted overall efficiency of 130 % results. The difference between the weighted overall efficiency of 130 % and the overall efficiency of 80 % corresponds to the power supplied by the ambient heat of 50 %, which is not evaluated as expenditure.

The same fuel can thus be used to produce 1,625 times the usable heat (130 % / 80 %). Alternatively, the same usable heat can be produced with 61.5 % of the fuel (1/1,625), which corresponds to a saving of primary energy of 38.5 %. The example described in Figure 2.11 is also entered as an example in Figure 2.10.

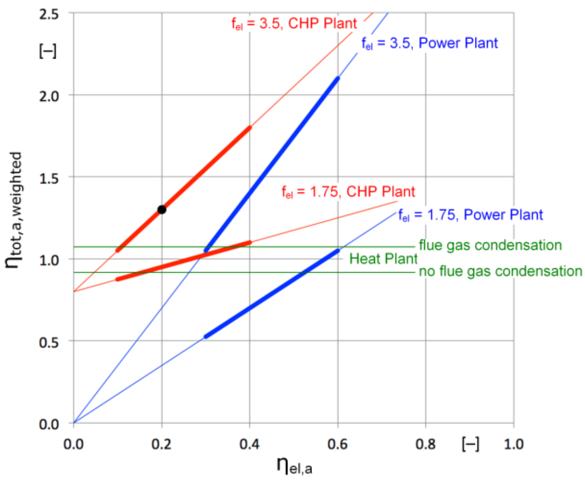


Figure 2.10 Weighted overall efficiency for: f_{el} = 1.75 and f_{el} = 3.5 for four scenarios: 1. Heating plant with or without exhaust gas condensation, 2. Cogeneration plant operated by heat η_{tot} = 0.8, 3. Power plant (Without waste heat utilization). The black dot at the graph corresponds to Figure 2.11.

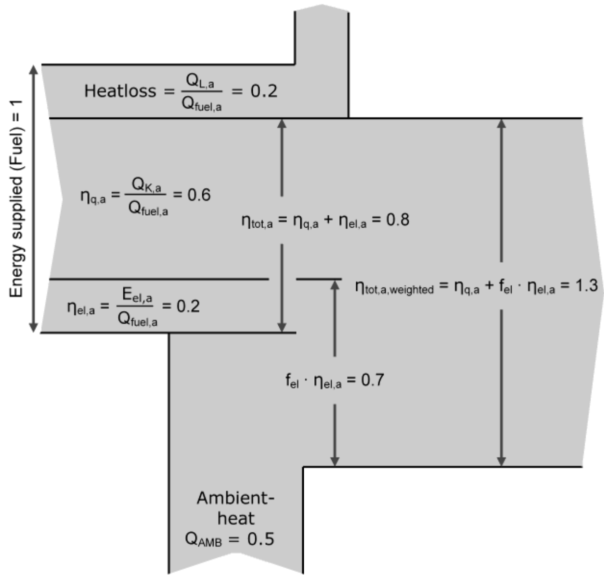


Figure 2.11 Sankey diagram for the scenario of a CHP plant whose electricity production is used to drive heat pumps. Assumptions: CHP : $\eta_{el,a}$ = 20 %, $\eta_{q,a}$ = 60 %. Weighting Electricity: f_{el} = Annual Performance Factor (APF) = 3.5. For this example, the weighted total efficiency is 130 % which is shown as a dot in Figure 2.10.

2.7 Heat generation and potential of district heating

A central plant for the production of heat is called a heating plant or, in the case of a district heating network, a district heating plant. If electricity and useful heat are produced, the central heating plant is called a combined heat and power plant (CHP) or co-generation plant. The following processes and energy sources are used as heat sources for district heating:

- Automatically operated boilers for energy wood such as forest wood chips, residual wood, waste wood and wood pellets. Corresponding wood boilers are used in district heating networks to cover the base load or the entire heat demand.
- Boilers powered by natural gas or fuel oil are often used to cover the peak load, while for new district heating networks a purely fossil fuel supply is excluded.
- Waste heat at a directly usable temperature level of over 70 °C from industrial processes can theoretically be used for district heating, but other measures such as internal processing are are favored.
- Waste heat from plants for combined heat and power generation at over 70 °C. Examples are steam turbines in waste incineration plants or wood-fired combined heat and power plants, wood-fired organic Rankine systems or block heat and CHP plants powered by natural gas or biogas. As CHP plants have high capital costs, they must be operated full time. CHP plants therefore often have a peak load boiler, while the waste heat used as district heating is used to cover the base load.
- Heat pumps for ambient or low-temperature waste heat. Ambient heat is from the air or from geothermal sources. Low-temperature waste heat is generated in waste water, for example, which can be used as a heat source in individual buildings or centrally in the sewage system. A large potential of low-temperature waste heat is treated water from sewage treatment plants (STP).
- Solar thermal systems can be used as auxiliary heating in district heating networks. In this case, a heat storage tank designed for one day is usually integrated. Solar power is designed for the summer hot water demand to lower costs. For existing residential buildings, solar power covers less than 10 % of the total heat generation whereas significantly

higher values are possible for new buildings. However, the integration of solar thermal systems in district heating plants is still not widespread in Switzerland, as the heat generation costs are still higher than the marginal costs.

- In regions with hot near-surface rock layers, direct geothermal district heating can theoretically be considered. In Switzerland, however, these are not common which is why near-surface geothermal energy has so far been largely limited to the provision of low temperature heat for heat pumps. If, on the other hand, deep geothermal energy and higher temperature levels are to be used, the focus is on electricity generation in a thermal process, whereby the waste heat can be used for district heating.

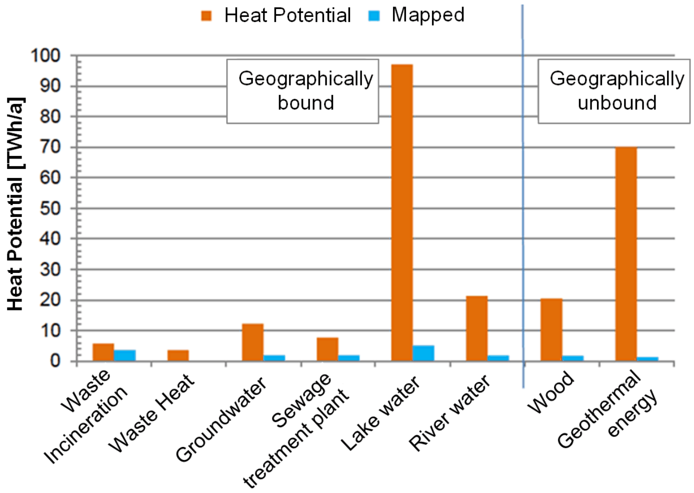


Figure 2.12 Heat potential of various heat sources (orange) and the "assigned" potential that can be distributed in 2050 with district heating at a maximum of 4.5 cent/kWh [19].

For Switzerland, the potential of the most important heat sources for district heating networks in 2014 was surveyed and a geographical division into clusters of 100 by 100 metres (hectares) was made. The potentials of MWIP waste heat, industrial waste heat, groundwater, sewage treatment plant waste heat, lakes, rivers, wood energy and geothermal energy were recorded. The heat potential is shown in orange in Figure 2.12. It does not include the potential of solar thermal energy and also does not take into account ambient air as a heat source, as this is mainly of interest for decentralised heat pumps.

Assuming that by 2050 the heat consumption of buildings will be reduced by 50% and that of industry by 20%, the potential for meeting the remaining demand has been estimated and the potentials "allocated" to district heating. These are based on the assumption that heat distribution generates costs of a maximum of 45 CHF/MWh, which is considered to be economically attractive. Figure 2.12 shows these allocated potentials in blue, which are broken down in detail in Table 2.3.

The survey shows that lake water and geothermal energy have by far the greatest potential. Among the potentials allocated, lake water is the most interesting with 29 %, followed by waste incineration with 21 %. With the exception of industrial waste heat, for which a value of zero is shown due to an insufficient basis for allocation, all other sources have potentials of 10 % to 11 % of the district heating potential.

The total allocated district heating potential is 17 TWh/a with a forecast demand of 45 TWh/a for 2050 and can thus cover up to 38 % of the total heat demand for space heating and hot water. Of the allocated potential, 69 % is accounted for by low-temperature sources for district heating using heat pumps. It should be noted that exploiting this potential requires a corresponding expansion of electricity production.

Table 2.3 Distribution of the allocated heat sources with a distinction between direct use and use by means of heat pumps, figures according to [19]. *The value for waste heat is shown as zero, as the basis for a geographical allocation is missing

How	Source	Heat Source	TWh/a	%
Direct	Wood	Wood	1.7	10
	Waste Heat	Waste (Incineration Plant)	3.6	21
		Waste Heat* (Industrial)	0	0
Heat Pump	Waste Heat	Sewage Treatment Plant (STP)	1.9	11
	Environment	Lakes	5.1	29
		Rivers	1.8	10
		Ground water	1.9	11
		Geothermal energy	1.7	10
Total			**17.0**	**100**
Overall direct utilization			5.3	31
Overall utilization with heat pumps			11.7	69
Overall demand for hot water and space heating in 2050			45	100
Geographically assigned potential district heat			17	38

2.8 Heat generation

2.8.1 Wood boiler

Although only a few years ago fossil fuel-fired boilers were still used for district heating in isolated cases, this is in Switzerland no longer allowed for new plants due to climate obligations. For this reason, biomass-fired boilers are often used to replace existing fossil fuel-fired systems and for the construction of new district heating networks. Until now, mainly energy wood was used because the supply of wood is stable and wood has the most advantageous characteristics as a biomass fuel. However, since energy wood can only replace part of the heat produced with fossil fuels, the interest in other biomass fuels such as residues from agriculture and food processing will also increase once the potential is in full use.

The technologies and supply chains for wood are already well established, with forest chips, residual wood and waste wood being used primarily for district heating, and wood pellets also being considered [24]. However, different product ranges require completely different techniques for storage, transport and combustion, and different construction and legal requirements also apply. The latter applies in particular to the pollutant emissions in exhaust gas regulated in the Clean Air Regulations (APCA or “Luftreinhalte-Verordnung LRV” in Switzerland) [25] and the disposal of ashes regulated in the Waste Regulations [26].

Together, all ranges of energy wood today contribute around 37'200 TJ/a or about 4.1 % to the total energy consumption of Switzerland. This could be increased by about 50 %, which would mean that wood would cover about 6 % of today's consumption or more, if consumption were to be reduced in the future.

In the case of **fresh wood chips** from the forest, the high water content, among other things, must be taken into account when designing the system. The use of wet forest chips requires suitable firing systems and is only possible from a certain minimum size. This also limits the possibility of rapid load changes and partial load operation. Firing systems with wet forest chips are therefore particularly suitable for covering the base load. When storing wet wood chips, carbon dioxide (CO_2) can be formed as a fermentation gas. Because this gas sinks due to its higher density than air, safety measures such as ventilation must be taken. The restrictions mentioned for forest chips also apply to wet residual wood, such as bark from sawmills. However, bark also has a high ash content and irregular pieces, which places additional demands on conveyor systems and incineration plants.

Depending on the application, **residual wood** from the woodworking industry, such as joineries, carpentries, furniture and kitchen factories, may contain additives such as glues and paints. Furthermore, the chunk size can vary from coarse pieces to fine shavings, which must also be taken into account. The handling and combustion of dusty fuels is particularly demanding, as they are associated with explosion hazards and personal injury and require special combustion systems. For the use of combustible dusts, therefore, pelletisation or briquetting is often used on location of production.

In the case of **waste wood**, i.e. wood from old furniture, packaging material and demolition of buildings, the origin and composition are usually not known, which is why waste wood is generally regarded as contaminated with heavy metals and other foreign substances. Waste wood may therefore only be used in plants that have an air pollution control act APCA permit and comply with the increased requirements for waste gas purification and plant operation.

Although this means that technically specialised plants for all types of wood exist, the specific care regarding storage, combustion technology and plant operation have to be taken into account, as described in QM Holzheizwerke [21]. For example, underfeed firing systems are often considered for fuels with low ash and moderate water content, while for increased ash and/or high-water content, pusher grate firing systems as shown in Figure 2.13 are usually used.

Underfeed furnaces produce up to 1 MW, grate furnaces, however, up to more than 20 MW. For capacities of 10 MW and above, fluidised bed furnaces are also considered. These achieve somewhat higher efficiencies and lower raw gas emissions with a somewhat higher auxiliary energy consumption.

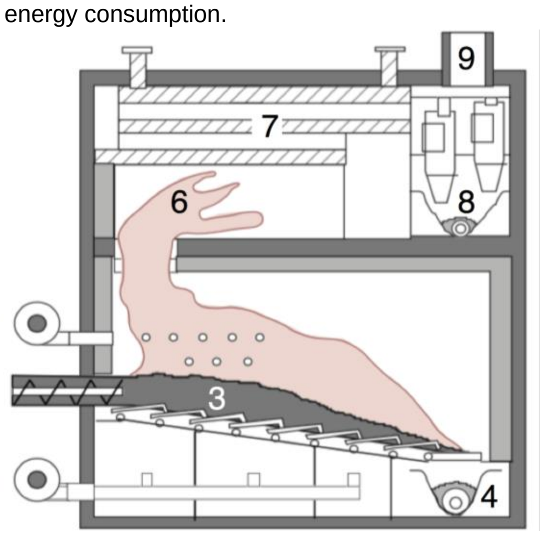


Figure 2.13 Principle of a moving grate furnace for wood chips and other wood types [27]. 1 Fuel supply, 2 Primary air, 3 Moving grate with fuel bed, 4 Discharge of grate ash, 5 Secondary air, 6 Afterburner chamber, 7 Boiler, 8 Multicyclone as preseparator of coarse dust, 9 Flue gas.

Automatic wood heating systems are equipped with dust separators to comply with the emission limits for fine dust. When using moist fuels, electrostatic precipitators are usually used, for dry fuels also fabric filters.

As can be seen from the layout of a wood-fired heating plant with fuel storage, boiler room and technical equipment (Figure 2.15), heating plants using wood require a large amount of space. This is also associated with high capital costs. For economic reasons, automatic wood-fired heating systems therefore usually aim for a high number of full operating hours by covering the base load.

The capital costs of wood-fired heating plants show strong economies of scale, especially for capacities below 1 MW to 2 MW, as shown in Figure 2.14. A major cost factor here is fuel storage. Storage halls are usually the cheapest, but in new buildings, underfloor silos can also be integrated at only slightly higher costs. However, subsequently added underfloor silos cause high costs. Due to the large amount of space required for fuel storage, it is important to have a secure supply all year round. Storage capacity should be designed for the coldest period of the year. The disposal of the ash must also be ensured. Often ash taken back by the wood supplier and disposed of correctly.

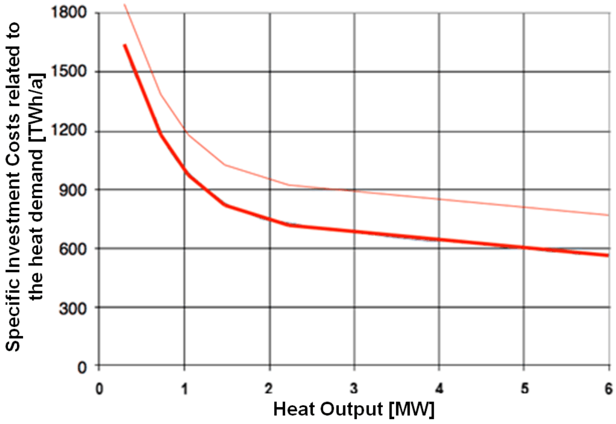


Figure 2.14 Specific investment costs for automatic wood heating plants (including fuel storage and boiler room) as a function of heat output. The low values apply to favourable structural conditions for fuel storage and boiler room.

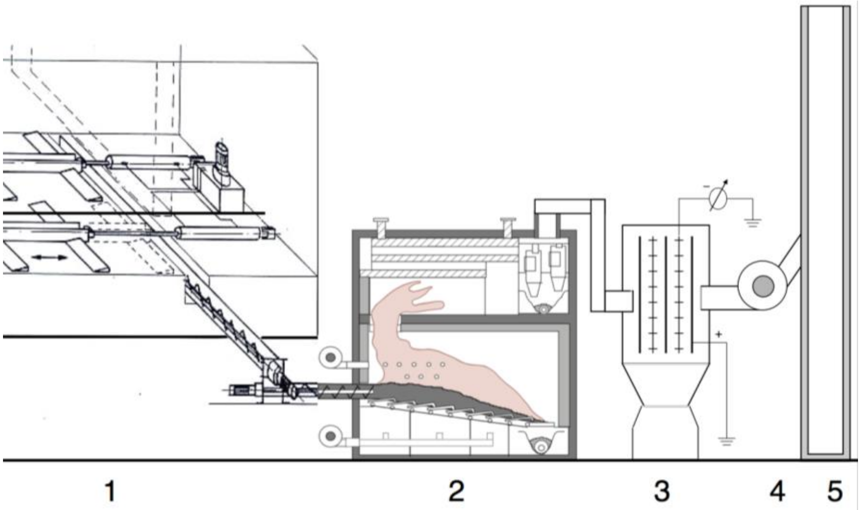


Figure 2.15 Heating system with automatic wood firing: 1 Fuel store with discharge, 2 Boilers, 3 Fine dust separators (in the figure designed as electrostatic precipitators), 4 Exhaust gas ventilator 5 Chimney [27].

Table 2.4 Calorific value and energy density for storage for different fuels. A typical water content (mass of water per mass of wet wood) of 30% is assumed for wood chips. The storage density refers to the mass per storage volume, which corresponds to the bulk density for wood chips. The energy density describes the energy content per storage volume.

Fuel		Water Content Wt.-%	Calorific Value kWh/kg	Storage Density kg/m³	Energy Density kWh/m³	Storage Volume (Fuel oil=1)
Wood Chips	Spruce	30 %	3.5	200	700	13
	Beech	30 %	3.3	270	900	11
Wood Pellets		10 %	4.9	650	3200	3
Fuel Oil		0 %	11.8	850	10'000	1

As an alternative to wood chips, **wood pellets** can also be used to generate heat. Wood pellets require only about a quarter of the space needed for the storage of wood chips, as the comparison in Table 2.4 shows. Although the space requirement is still greater than for fuel oil, a pellet heating system is often more suitable than a wood chip one. During storage, it should be noted that wood pellets release carbon monoxide (CO) and can thus lead to a risk of suffocation in storage rooms above a certain size. Ventilation and monitoring are therefore necessary for larger storage rooms. However, there are some different requirements than for wood chips. Since wood pellets have a low moisture content and homogeneous properties, they are also suitable for use in small furnaces but due to the production effort involved, they are more expensive. Therefore, other types of energy wood are generally used for base load coverage in district heating plants. However, in order to completely replace fossil fuels, it is conceivable in future that pellet boilers could be used instead of oil or gas boilers for summer operation and peak loads.

Because wet wood chips can lead to unfavourable operating behaviour in small furnaces and are not always suitable for summer operation in larger plants, wood chips with a reduced moisture and ash content have been offered for some years. Such"**quality chips**" are produced by technical drying and sieving. According to FAQ 36, QM Holzheizwerke differentiates between fine and coarse chips, whereby the coarse category is only recommended for systems from 100 kW [28].

As sustainably renewable wood in Switzerland only allows an expansion of about 50 % to a maximum of 100 %, there is an increasing interest in other **biogenic fuels**. These include, for example, agricultural and food processing residues such as from straw and grain. These have a high ash content as well as high contents of nitrogen, sulphur, potassium, chlorine and other substances. This results in large quantities of ash, increased slagging and deposits as well as increased emissions of dust and nitrogen oxides. The use as fuel therefore causes higher demands on combustion technology and plant operation. Meat-and-bone meal and dried sewage sludge are now also available, whereby the disposal requires strict hygiene and security measures. These resources should be used in the future for energy production; however, the building of appropriate plants demands special attention. Therefore, investments in this area are usually limited to the food industry or waste management technology.

Wet biomass such as liquid manure from agriculture, sewage sludge from waste water treatment and solid kitchen waste are not suitable for direct use in heating boilers. However, these residues can be used as a substrate for **anaerobic fermentation**. De-centralised biogas plants are used in agricultural enterprises, but are generally small due to their cost. Larger central fermentation plants are used, which are often designed for the co-fermentation of liquid residues such as sewage sludge and liquid manure together with solid residues from gastronomy and households. The fermentation gas, biogas or sewage gas from fermentation plants usually consists of around 60 % to 70 % methane and 25 % to 40 % carbon dioxide and also contains sulphur and nitrogen compounds, which are usually separated before use. Due to its composition, fermentation gas has a calorific value that corresponds to about two thirds of the calorific value of natural gas. Fermentation gas can therefore be used in a gas engine CHP or processed for feeding into the natural gas network. Contrary to the use in motors, carbon dioxide must first be separated in a gas wash. When used in a CHP, waste heat is used for long-distance heating and often serves for covering the base load.

2.8.2 Directly usable waste heat

Industrial waste heat at temperatures above 70 °C is to be used or recuperated within the factory in the first priority and is only available for district heating if no suitable operational use is possible. Waste heat utilisation must be affordable, the location suitable and production viable. Heat supply is usually guaranteed by conventional heat generation. It should be noted that improvements or changes in production can reduce the amount of waste heat generated. In addition, much shorter depreciation periods are assumed for industrial production than for district heating networks, which makes it more difficult to coordinate investments. For these reasons, direct use of district heating from industrial waste heat is rare. If, on the other hand, large quantities of waste heat are available at temperatures above 300 °C, electricity generation with combined heat and power is a viable option, which is discussed in Chapter 0.

2.8.3 Low temperature waste heat and ambient heat

2.8.3.1 Heat pumps for raising temperature

Heat pumps use heat at a low temperature level and raise the temperature to a usable level. In compression heat pumps this is done by supplying mechanical energy, in absorption and adsorption heat pumps by supplying heat at high temperature. The ratio between useful heat and high-quality energy, usually supplied as electricity, is defined as the coefficient of performance (COP) according to Chapter 2.6.3 and, when considered over a year, is referred to as the annual performance factor (APF).

The coefficient of performance becomes infinitely large when the temperature difference between the heat source (lower temperature T_{low}) and heat production (upper temperature T_{up}) approaches zero. If the temperature difference becomes very large, the coefficient of performance approaches one, which turns an electric heat pump into an electric heater. Heat pumps are therefore only interesting if the source and useful heat have a small temperature difference. Figure 2.16 shows the Carnot COP as a function of temperature for different sources.

This assumes that low temperature usable heat is sufficient or that the heat pump is only used for pre-heating, such as raising the return temperature, and that post-heating is provided by a second heat generator. In addition, the heat production can also be designed for two temperature levels. To supply buildings, for example, a heat pump can alternately provide room heating at 35 °C and hot water at 60 °C. Without reheating, heat pumps for district heating networks are generally only suitable for supply temperatures up to a maximum of 70 °C. At a source temperature of 10 °C and a quality grade of 0.5 this corresponds to a COP of 2.9, as shown in Table 2.5.

On the other hand, if the heat pump is only used to preheat to 50 °C, a COP of 4.0 is achieved; at 35 °C a value of 6.2.

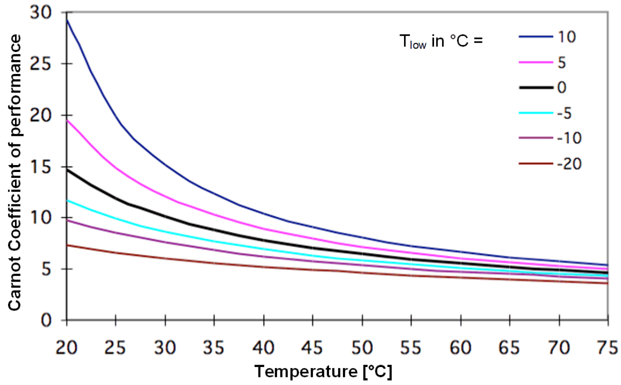


Figure 2.16 Carnot coefficient as a function of temperature T for different levels of the lower temperature (source temperature) T_{low}.

Table 2.5 Real coefficient of performance of a heat pump with quality grade 0.5 for different temperatures of the heat supply T_{SPLY} and the usable heat T.

	T_{SPLY} =	0 °C	10 °C	20 °C
T =	35 °C	4.4	6.2	10.3
	50 °C	3.2	4.0	5.4
	70 °C	2.5	2.9	3.4
	90 °C	2.0	2.3	2.6

In principle, a separate provision of space heating and hot water is also conceivable for district heating networks. However, this would require alternating network operation or a three-wire system, which would increase the operating costs and, in the second case, the investment costs.

For decentralised applications, compression heat pumps are mainly used, which have an electric drive for the compressor. For outputs of at least 300 kW, a direct mechanical drive is also possible, for example using a gas engine.

The design of a heat pump is identical to a refrigeration machine, whereby the heat extraction at a lower temperature level is used to provide cooling. If there is a simultaneous demand for heat and cold, such as when combining an indoor swimming pool and an artificial ice rink, both the cold and the heat can be used and the electricity consumption of a compression system can be reduced accordingly. The heat produced by the refrigeration machine serves as directly usable waste heat. If there is no simultaneous heat demand at the place of cold production, cooling machines can also be used as waste heat sources for district heating.

The use of absorption systems is of interest, for example, if large amounts of waste heat are available at a sufficient temperature and a corresponding cooling demand exists at the same time. Due to the complexity of the equipment, however, such applications are limited to larger units. If district heating is available at low cost and at a high temperature, decentralised cooling production using absorption systems is also a possibility

2.8.3.2 Heat sources for heat pumps

For central heat generation in district heating networks with heat pumps, the following heat sources are particularly suitable:

- **Ambient air**. Decentralised heat pumps are often designed with ambient air as the heat source, as this application is possible almost everywhere and the drilling costs for individual probes are relatively expensive. However, ambient air has the lowest temperature in the cold season with maximum heating demand, which limits the performance figures in winter. For this reason, the combination of a heat pump with a second heat generator is an option. In addition, the efficiency of air heat pumps can be increased by preheating air, for example in geothermal heat exchangers. For district heating plants, however, the focus is on heat sources such as geothermal energy, water bodies and waste heat.
- **Near-surface geothermal energy**. From a depth of 10 m, the ground has a temperature of approximately 11 °C, which remains almost constant throughout the year. It increases by around 1 °C per 30 m depths. Geothermal probes powered by brine take advantage of this heat. In Switzerland, the drilling depths are usually between 100 m and 300 m and the source temperatures from 12 °C to 15 °C, which allows higher coefficients of performance in winter than with ambient air. The sustainable heat extraction from a geothermal probe over a long period of time depends on numerous factors such as drilling depth, rock layer, probe length and probe density. Especially for larger units, it is therefore advisable to use a probe field to dissipate excess heat from buildings and solar systems in summer. In this way the soil can be regenerated and the probe field can be used as a seasonal storage tank. At typical probe temperatures, cooling of buildings with active components is possible without air conditioning. The use of hybrid solar thermal systems with actively cooled photovoltaic systems is also possible. A further application is unglazed solar collectors, which are used in summer for free cooling of the building at night and during the day dissipate a considerable amount of excess heat to the probe array. In addition to geothermal probes, ground collectors at depths of around 1.5 metres are an option. However, since these require large areas of land and achieve lower temperatures than geothermal probes, they are of little significance.
- **Bodies of water**. The use of heat from groundwater, lake water and rivers are interesting from a temperature point of view. While groundwater requires an extraction well and an absorption well, two pipes are sufficient for the use of lake water. Groundwater use is seen as a kind of geothermal energy and is therefore described further under Geothermal Energy. Since many towns and villages are located on water, lakes and rivers offer the greatest potential for heat pumps. Corresponding applications are also partly supported by the public sector. For the use of natu-

ral water bodies, requirements regarding water protection and nature conservation must be observed. For example, water may only be cooled by a maximum of 3 K according to a guideline value set by the Federal Office for the Environment. In winter, however, the first 10 m of lake water has typical temperatures of over 5 °C; at 20 metres, between 10 °C and 15 °C. Even when the temperature reduction of 3 K is fully exploited, this allows high coefficients of performance. As water temperatures remain largely below 18 °C in summer from 20 metres upwards, lake water can also be used for cooling buildings through free-cooling. The impact on the temperature of the water bodies remain very low. An added benefit is that the slight cooling counteracts the undesired warming as a result of climate change.

- **Sewage**. Wastewater from households, public institutions and industry has a temperature of 20 °C to 25 °C, which is significantly higher than drinking water, which is supplied at 8 °C to 12 °C [29]. This waste heat in wastewater can be recovered in various ways. In principle, heat recovery can be used directly at the consumer to heat up fresh water thus reduce decentralised hot water consumption. Corresponding equipment is available, for example, in the form of shower trays with built-in heat exchangers. This undiluted waste water has an even higher temperature of over 30 °C. Because operating times of these individual facilities in residential buildings are short, heat recovery from the mixed waste water from several consumers can be considered as an alternative. At temperatures of 20 °C to 25 °C the waste heat is a suitable source for heat pumps. Such applications can be carried out by heat exchangers of individual buildings as shown in Figure 2.17 "In-house", although for economic reasons this is only possible for larger units from about 150 kW upwards.
  In sewer systems with a daily minimum discharge of at least 10 litres per second, which corresponds to about 5000 inhabitants, the use of wastewater collected in sewers is also possible. Figure 2.18 shows a heat exchanger introduced into the sewer system. Alternatively, a separate shaft with heat extraction can be installed. While decentralised applications reduce the demand of individual buildings, heat recovery from the sewer system can be used to supply district heating networks. It should be noted that the heat potential from the sewer is reduced by decentralised heat recuperation and therefore appropriate coordination is required.

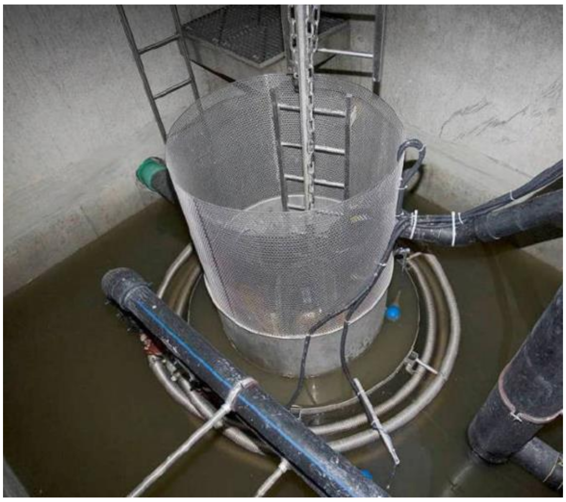


Figure 2.17 Heat Recovery of Waste Water by In-house-device [30].

- **Waste water treatment plants** (WWTPs). A centralized use of waste water heat is suitable for waste water treatment plants. The use of the treated water in the outlet of the sewage treatment plant is of particular interest here. With typical water temperatures in winter of over 8 °C, cooling by at least 4 K is usually possible. The potential corresponds to a heat output of about 800 MW [30]. Sewage gas is often used in a CHP for power generation, and any waste heat from motors can be used to supply district heating.

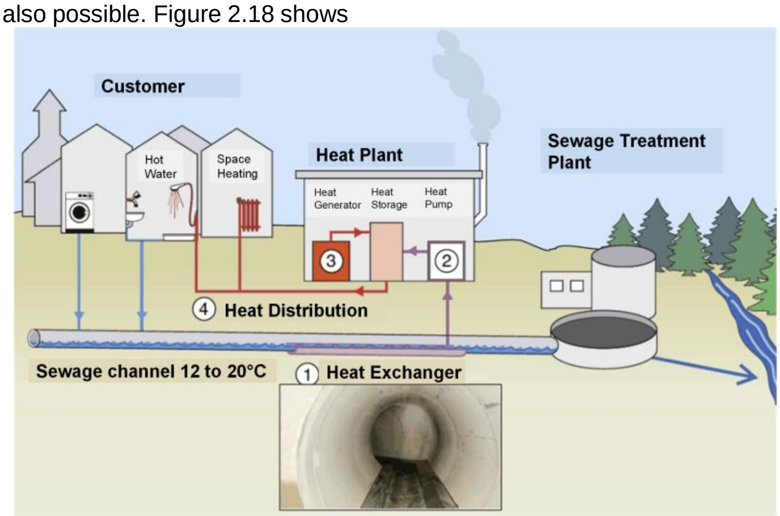


Figure 2.18 Waste Heat Utilization from the Sewer System [31]. 1 Heat Exchanger, 2 Heat Pump, 3 Heating Generator for Peak Load, 4 Heat distribution with the District Heating Network.

2.8.4 Solar thermal energy

Solar thermal energy uses the sun's radiation, which at midday, when the sun is high and the sky is clear, amounts to around 1000 watts per square meter on a horizontal surface. An irradiation of 1000 W/m2 also serves as a reference value of the performance of solar thermal systems. On an annual average, an effective value of around 125 W/m2 or 1/8 of the radiation occurring at a cloudless midday is achieved in Switzerland. The average radiation on a horizontal collector thus corresponds to 1/8 x 24 h or 3 h per day and 1/8 x 8760 h/a or 1100 full operating hours per year, which in turn corresponds to 1100 kWh per square metre and year. Since Europe is located north of the Tropic of Cancer, the angle between the sun and the earth's surface is less than 90° even on the longest day when most radiation comes from the south. For this reason, permanently installed collectors achieve the maximum annual yield at an inclination of around 42° to the south. However, since the winter yield is decisive for solar thermal energy and the sun then hits the earth's surface at a flat angle, the maximum winter yield is achieved with an inclination of around 60°. It is also interesting to install the system on a vertical south wall, as its winter yield is only slightly lower than at an inclination of 60°. In order to prolong the daytime yield, an east and west orientation at a flat inclination is ideal, which is mainly used for photovoltaic systems. In all cases, the fact that the yield is only generated during the day means that solar thermal systems are equipped with a heat storage tank.

Because part of the radiation is reflected (which results in optical losses) and the collector also shows increasing thermal losses to the environment as the temperature rises, solar thermal systems for hot water production achieve typical efficiencies of around 60%, as shown in Figure 2.19 using the example of the collector characteristic curve of a flat-plate collector operated at an ambient temperature of 5 °C and irradiation with 1000 W/m2 . With reduced irradiation, the stagnation temperature achieved without flow decreases and thus also the efficiency.

The system efficiency evaluated over one year includes additional losses due to storage and system integration and reaches typical values of around 50% if the system is designed for a solar coverage of around 50%, as Figure 2.20 shows. For such plants, a yield of 550 kWh per year and square meter of absorber surface can be achieved in the Swiss midlands. Even slightly higher yields per square meter of absorber surface are achieved with vacuum tube collectors, which also achieve higher temperatures due to better thermal insulation.

In residential buildings, solar thermal systems are often designed in such a way that the hot water supply can be fully or partially covered during around three to four summer months. As the example in Figure 2.20 shows, relatively small collector surfaces of 1 m^2 to 1.5 m^2 per person are sufficient for this. Because the yield in the winter months falls to below 25% of demand, additional heating is required. A solar coverage rate for hot water of 50% to 60% is thus achieved over the year.

The degree of coverage can be increased by enlarging the collector surface. However, this increases the losses as a result of the higher temperatures and the surplus heat that cannot be used in summer, which is why the system efficiency and the solar yield per square meter decreases. On the contrary, higher system efficiencies and yields of up to 700 kWh per square metre and year are achieved if solar heat is used only for preheating with a degree of coverage of 5 % to 30 %, as shown in Figure 2.21. Solar thermal energy is therefore also economically interesting for integration into district heating networks, especially for preheating or for covering the summer hot water demand.

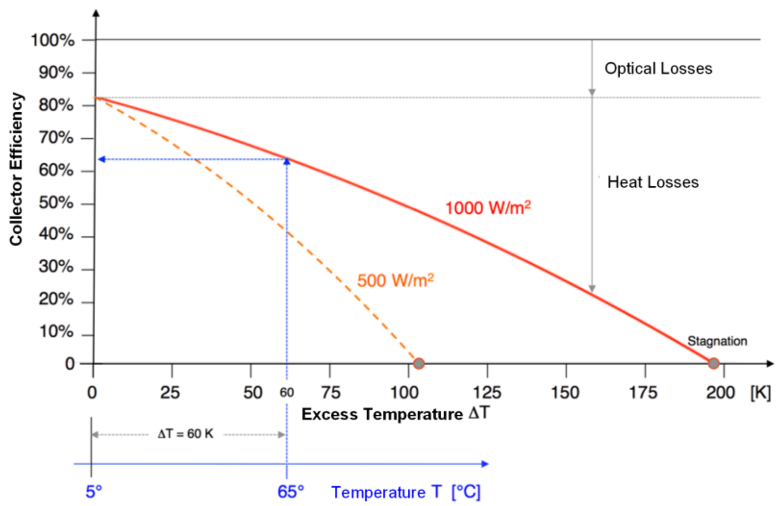


Figure 2.19 Collector efficiency as a function of the excess temperature (= collector characteristic curve) of a glazed flat plate collector (difference between collector and ambient temperature).

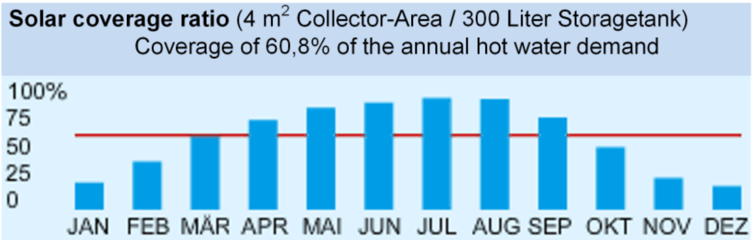


Figure 2.20 Monthly solar coverage ratio for hot water generation by a solar system specified for the summer demand of a house hold with 4 Persons and 4 m^2 collector area.

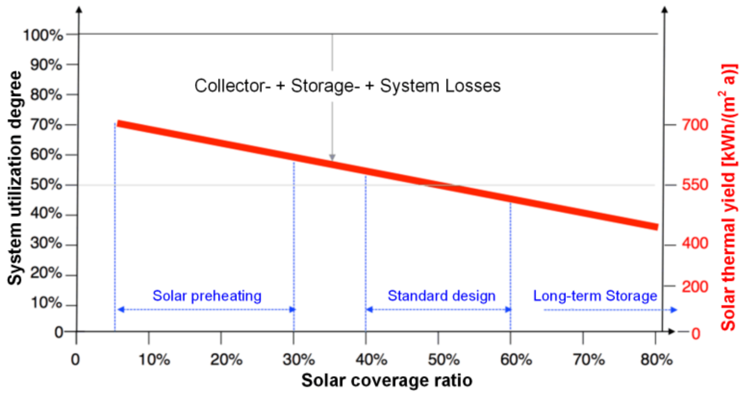


Figure 2.21 System utilization factor (left) and solar yield (right) as a function of the solar coverage ratio.

While short-term storage is sufficient for solar preheating, an increase in the heat storage capacity is necessary to increase the solar fraction, which is possible with geothermal probes. Thus, for example, the following concepts for solar-supported district heating can be considered:

1. Solar-supported district heating with short-term storage: In this concept, solar heat is used for preheating, which enables high solar efficiency, while excess heat is stored in a short-term storage tank. Because solar radiation and space heating requirements are seasonally generated in opposite directions, only a small proportion of the annual demand can be covered by solar energy using this principle, typically between 5 % and 12 % in today's networks. However, since the share of hot water in the annual demand will rise from 10% for old buildings to around 50% for new buildings, a share of solar heat of 25% or more is conceivable in future with short-term storage tanks.

2. The integration of solar heat with seasonal heat accumulators is possible in different ways. Solar heat can be used at a low temperature level and thus with high efficiency for heat in winter and in summer to regenerate the rock layers around geothermal probes. The heat is then supplied by heat pumps, which achieve high annual performance factors due to the high source temperature. Since low temperatures are sufficient to regenerate the rock layers, unglazed collectors are used for such applications. These only achieve high efficiencies at low excess temperatures and the stagnation temperature is much lower than in a glazed collector shown in Figure 2.19 due to the increased heat losses. Unglazed collectors can also be used for freecooling at night. In addition, langite heat storage tanks at a higher temperature level are also possible, whereby the consumers are supplied with reheating from the storage tank, for example via a wood boiler. Thus, solar coverage rates of up to over 40 % are possible.

3. In addition to solar thermal systems at the heating plant, it is also conceivable for future networks to integrate decentralised solar thermal systems into a district heating network, for example by raising the return flow.

Solar thermal systems have high investment costs, which decrease with the size of the system. It is only in larger units, such as for multi-family buildings, that heat production costs between 10 cent/kWh and 20 cent/kWh are achieved, as Figure 2.22 shows.

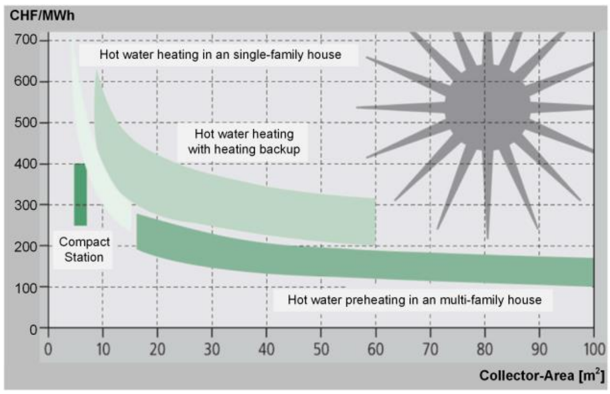


Figure 2.22 Heat production costs for solar hot water and heating support [32].

2.8.5 Geothermal energy

A distinction is made between near-surface and deep geothermal energy.

Near-surface geothermal energy (usually down to depths of about 400 m) includes geothermal probes, ground collectors and groundwater and pit water, which are usually used as a heat source for heat pumps.

For district heating networks, the use of groundwater for a heating capacity over 100 kW is interesting. In addition, geothermal probes have great potential. Figure 2.23 shows the arrangement for a field with 10 geothermal probes for heating a multi-family building.

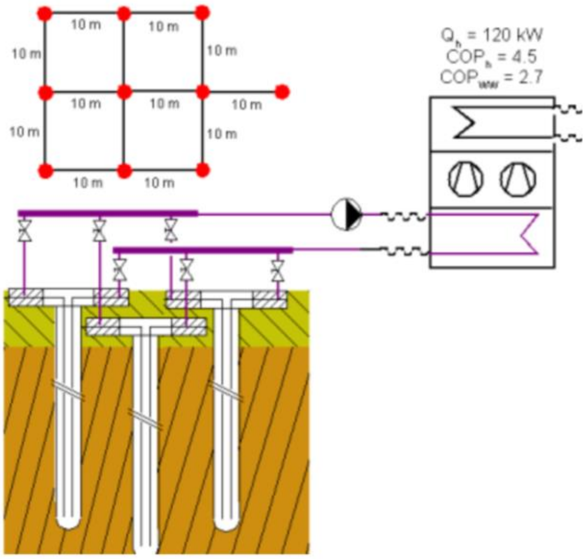


Figure 2.23 Example of a field with 10 heating probes with 210 m drilling depth to supply a multi-family house with a heating capacity of 120 kW with COP of 4.5 for space heating and 2.7 for hot water [33].

Because the temperature in the ground decreases due to heat extraction and the probes influencing each other, the transient behavior of the heat extraction must be taken into account for the design. To limit the temperature decrease, probe fields can be regenerated by active cooling of the building and by solar thermal energy.

For the design of probe fields, calculation programs for transient behaviour are available, which also allow the integration of active cooling and solar regeneration [33]. Figure 2.24 shows a calculation example for the behaviour of the probe temperature over 50 years.

Due to the large surface requirement of around 25 m^2 per kW, ground collectors are usually used for outputs of up to 10 kW for individual buildings and are therefore not common.

In the case of deep geothermal energy, a distinction is made between the hot dry rock process (HDR process), deep geothermal probes and hydrothermal geothermal energy. In the HDR process, the geothermal energy in the dry, hot rock is used by pressing water through an injection borehole into rock layers several kilometres deep. In the process, existing cracks are widened and, in some cases, new ones are created. The water circulates through this system, heats up and is then pumped to the surface via a second borehole (production borehole). The water can be used to generate electricity in ORC plants or steam turbines. The heat can also be used directly for district heating, but it is generally more advantageous to use the waste heat from electricity generation. However, the HDR process is only of interest in regions where the temperature rise in the rock exceeds 50 °C per kilometre depth. However, since previous attempts to use deep geothermal energy in Switzerland have triggered unexpected earthquakes or failed to meet yield expectations, this technology has not yet been of commercial significance

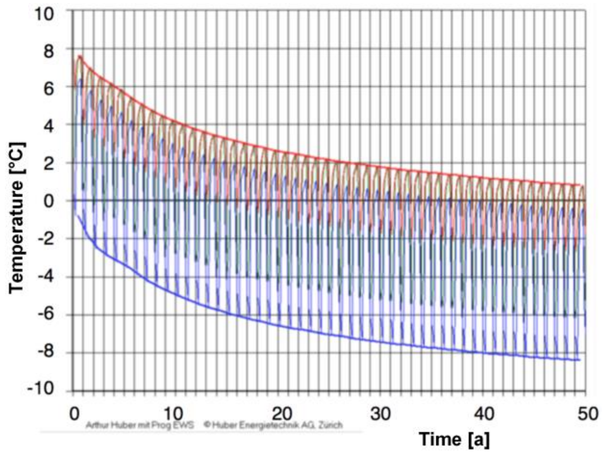


Figure 2.24 Temperature decrease of a probe field during 50 years [33]. Shown is the monthly maximum (red) and minimum (blue) of the probe inlet temperature.

2.9 Combined heat and power (CHP)

2.9.1 Overview

A combined heat and power plant is an energy centre in which power is generated in a thermal process and the waste heat produced is provided as useful heat. This principle is known as combined heat and power (CHP) and the power is usually used to generate electricity, as Figure 2.25 shows using the example of an internal combustion engine with waste heat recovery.

As CHP plants have high investment costs, they are usually supplemented by a boiler to cover peak loads. When using a single CHP unit, for example, the design is based on around 30 % of the maximum heat load demand, so that a peak load boiler is used for the demand above the bivalence point, as Figure 2.26 shows.

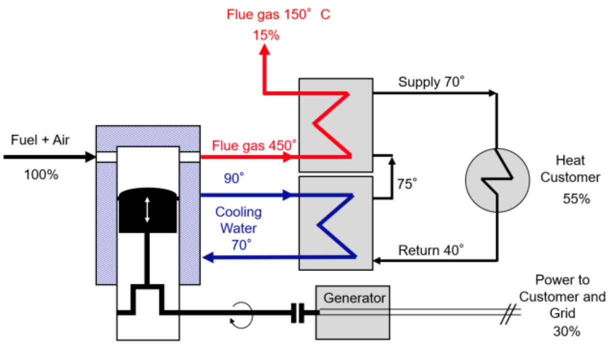


Figure 2.25 Principle of the combined heat and power cycle demonstrated by the example of a combustion engine.

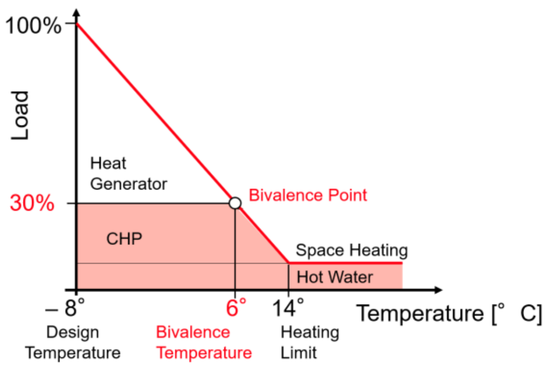


Figure 2.26 Design of a CHP unit.

2.9.2 Steam turbines

Steam turbines use water as the process medium in a closed cycle with evaporation and condensation, corresponding to the Rankine process as shown in Figure 2.27. When steam turbines are used for co-generation, they can be designed as backpressure turbines using the entire waste heat or as extraction condensing turbines with controllable waste heat utilisation. In contrast, a condensing turbine produces only electricity, while all the waste heat is dissipated into the environment, as in nuclear power plants.

The net electric efficiencies of steam turbines in the output range of typical wood-fired cogeneration plants or of waste incineration plants (1 MW_{el} to 50 MW_{el}) are between 10 % and 25 %. Due to the low electric efficiencies, heat operation is advisable. In the output range of coal-fired power plants with up to over 1 GW_e, electric efficiencies of up to over 45 % are possible without heat extraction. However, for such applications only partial heat utilisation is possible at most. It should be noted that the electric efficiency of steam power plants is reduced by heat extraction, as the usable heat is required at a temperature above ambient temperature.

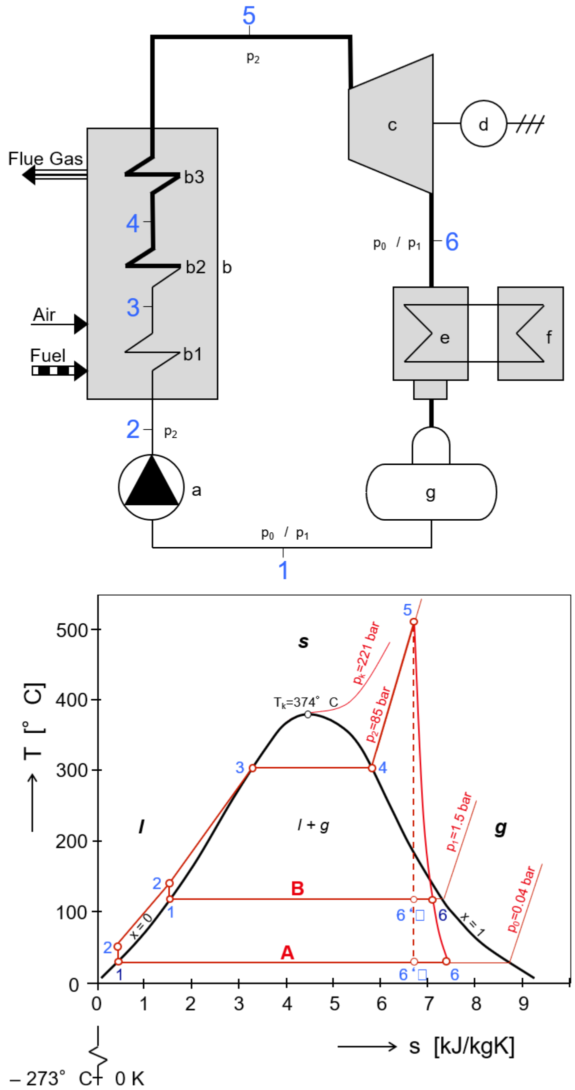


Figure 2.27 Structure of a steam turbine power plant (top) and Rankine process in the T,s slidegram. 1-2 Pressure increase of water by the feed pump a, 2-3 Heating of water to evaporation temperature in the boiler b, 3-4 Evaporation at constant temperature in the evaporator, 4-5 Superheating of the steam in the superheater, 5-6' Work output by frictionless expansion of the steam in the turbine c, 5-6 Work output with non-ideal turbine, 6-1 Condensation of the steam in the condenser e with heat output to the cooler f and return to collecting tank g.

Although steam plants can also be operated at partial load, the control range is smaller than with pure heat generators. Rapid load changes must be avoided and partial load operation leads to a significant drop in efficiency below a certain output. Where steam systems are used as heat generators in district heating plants, they are normally designed for base load and supplemented by peak boilers. The contribution of steam plants and applications of combined heat and power generation to electricity generation is therefore limited.

2.9.3 Steam engines

For small outputs (up to 1 MW_{el}), steam engines are also used based on the Rankine principle. Steam piston engines and screw-type steam engines are available. The net electric efficiencies are around 10 % to 12 %, which is why heat-guided operation is preferred.

2.9.4 Organic rankine cycle

In addition to water-operated steam turbines for small and medium capacities (up to 2 MW el), processes with organic media which boils at lower temperatures are also used. The process is then called the Organic Rankine Cycle (ORC). For applications with wood, this principle offers the advantage that a secondary circuit operated with a heat transfer oil is possible between the heat generator and the steam generator, thus simplifying plant operation. Since the auxiliary energy consumption is relatively high, the net electric efficiencies are between 10 % and 15 %. In addition to this the emission temperatures of boilers operated with thermal oil are higher than in water boilers which is why additional heat is usually extracted from the hot flue gas

2.9.5 Internal combustion engines

Internal combustion engines, which are mainly gas-powered, are used for minor stationary applications. These are operated using the four-stroke cycle with spark ignition, or, for smaller applications, with biogas and wood gas, sometimes also operated as ignition jet engines with the addition of liquid fuel as ignition aid.

The heat generated from exhaust gas (about 60 % of the waste heat) and from cooling water and lubricating oil (about 40 %) can be used for heat according to Figure 2.25. Cooling water and lubricating oil usually have temperatures of 80 °C to 90 °C, and in hot-cooled engines even over 100 °C. The temperature level of the exhaust gases is considerably higher at over 350 °C and depends on the engine type.

The use of an engine for co-generation is also called a combined heat and power unit (CHP). Biogas from fermentation plants and in some cases product gas from a wood gasification plant is used as fuel. In order to protect the engine from impurities and at the same time reduce pollutant emissions, gas purification is required in both cases. While high reliability can be achieved with biogas plants, the operation of wood gasification plants and the necessary gas purification is complex. Sometimes co-generation plants powered by natural gas are also used as base load heat generators. This enables reliable operation, but only a small proportion of natural gas is permitted in largely CO_2-free district heating.

Gas engines typically achieve efficiencies of 25 % for outputs below 50 kW_{el} and up to 35 % from 100 kW_{el} and over 40 % from 1 MW_{el}. When operating with wood gas, the efficiency of wood gasification is around 75%, which results in net efficiencies of 15% to 25% typical for wood gas CHPs. The same applies to biogas, where fermentation achieves a much lower degree of efficiency, but is used primarily to process organic residues.

In order to increase electric efficiencies, it is possible to use the waste gas heat in a combined process to generate electricity in an external thermal process, for which ORC systems are primarily suitable. Engines with integrated ORC systems are not yet widely available. However, in order to comply with increasingly strict fuel consumption regulations, especially for heavy-duty traffic, developments are underway to use waste heat recovery with ORC modules for diesel-powered series engines. If corresponding series-production units become available, this may enable the use of cost-effective ORC modules for CHP waste heat.

2.9.6 Open gas turbines

The same principle as for engines can be used with open gas turbines for higher outputs. In contrast to engines, the waste heat is generated almost exclusively in the exhaust gas at temperatures between 400 °C and 600 °C. Although this waste heat can basically be used for district heating, its use for driving a downstream steam process is more sensible. If the output is large enough, a steam turbine is used for this purpose, and the principle is then referred to as a gas and steam combined cycle power plant (CCGT). Combined cycle power plants powered by natural gas typically have a total output of 600 MWel and achieve electrical efficiencies of 60 %. In principle, further waste heat utilisation is also possible in combined cycle power plants but this is of secondary importance.

Gas turbines can also be used in the power range of gas engines, for example as a biogas CHP. The main advantage here are reduced costs for service and maintenance. In contrast, small gas turbines, also known as micro gas turbines, achieve significantly lower efficiencies than gas engines.

2.9.7 CHP and heat pump

Mechanically driven heat pumps with a combustion engine are a good option for larger outputs. Here, waste heat is available at the high and medium motor temperatures while the mechanical power is used to drive the heat pump at a lower temperature level. Corresponding applications are of interest, for example, if the waste heat of a biogas CHP is not sufficient for heat supply or if there is a simultaneous cooling demand which can be covered by the cold side of the heat pump.

2.9.8 Closed gas processes

In internal combustion engines and open gas turbines, fuel is burned directly in the engine where there are possible fuel impurities. Steam power plants, on the other hand, have an external heat supply by means of heat transfer, with a fuel or other sources such as geothermal or solar thermal energy serving as the heat source. Substances that are critical for the machine, such as chlorine in municipal waste, potassium in wood or sulphur in biogas, for example, cannot damage a steam turbine. However, if a fuel is produced from biomass or waste, the critical substances have to be removed in a complex gas purification process beforehand. Steam power plants have so far been used primarily for wood-fired power plants, which have high specific investment costs and low efficiencies for a small output. For this reason, other processes which are also operated via external combustion with heat transfer, but are less complex, are in demand. Stirling engines and closed (i.e. externally fired) gas turbines are used for this purpose. The mechanical design of these machines is comparable to that of combustion engines and open gas turbines, but the process medium is a gas such as air or helium which is circulated and heated externally. Stirling engines are used for small outputs from 1 kW up to around 100 kW and achieve higher efficiencies in this power class than closed processes. Closed gas turbines are used for outputs from a few kW up to over 100 kW. For both technologies there are numerous developments for applications with biomass fuels and concentrated solar energy, some of which are in operation in pilot plants. To date, these plants also have very high investment costs and therefore have few commercial applications.

2.9.9 System comparison

As Figure 2.28 shows, cogeneration technologies cover a range from a few kW to over 1 GW of electrical power. The net efficiencies that can be achieved range from less than 10 % to 60 %. With the established processes of steam power technology, the electrical efficiencies show a pronounced scale dependence of less than 10 % for outputs from 10 kW up to around 45 %. Alternative technologies such as wood fermentation and Stirling engines achieve higher efficiencies in the small and medium power range.

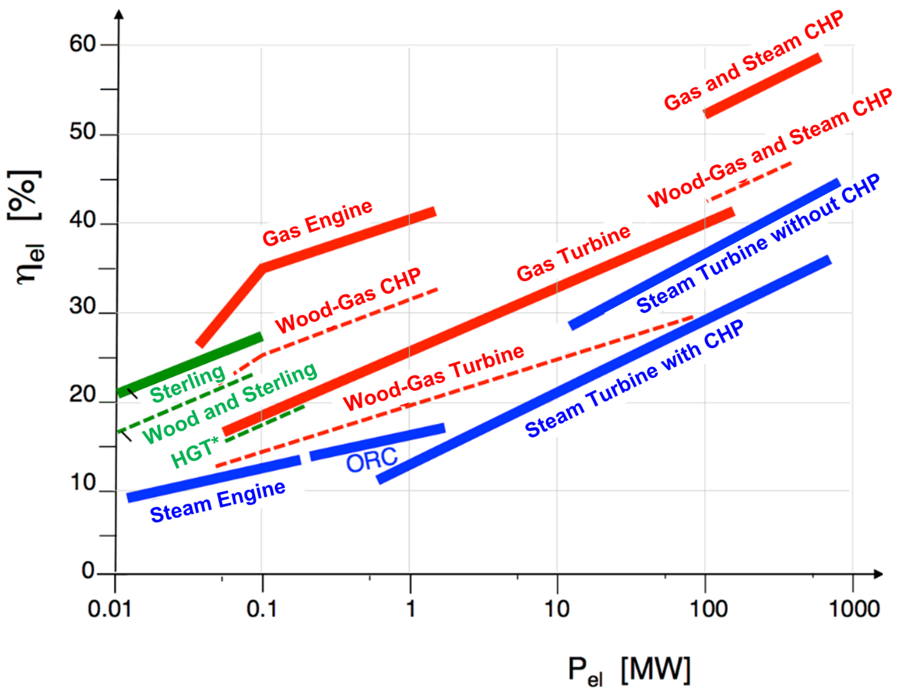


Figure 2.28 Net-Efficiency of various power generation technologies in functionality of the electric power. *HGT = Hot Gas Turbine (with wood combustion).

2.10 Thermal energy storage

The fundamental task of a thermal energy storage (TES) system is the temporal decoupling of heat generation from heat release. In this way, load peaks as well as load reductions, for example, can be well compensated for. Depending on the dimensioning of the energy storage, shorter or longer periods can be bridged. With combined heat and power generation, plant operation can be made more flexible and heat generation dimensioning smaller.

Depending on the system, the use of thermal energy storage therefore offers a considerable potential for optimising plant operation in terms of costs and primary energy consumption. The following points deal with the basics of thermal energy storage, design and application or integration. The information is largely based on the sources [34] and [35].

2.10.1 Properties and terms

The thermal energy storage systems can be differentiated as follows based on their areas of application and properties:

- **Temperature level**: The temperature at which a heat storage system is charged and discharged determines its area of application. For district heating applications, the temperature range is usually between 25 °C for surface heating systems such as underfloor heating and up to 90 °C for radiator heating. For hot water, temperatures of over 60 °C are necessary for hygienic reasons. For very extensive, large district heating networks, supply temperatures of over 100 °C are required. For industrial processes, such as the supply of steam, temperatures of over 100 °C (up to 250 °C) are required. High-temperature applications of storage tanks at temperatures of over 300 °C to 600 °C are used in solar thermal power plants, for example. Cold accumulators for cooling and air-conditioning are required in a temperature range of 5 °C to 18 °C. For industrial cooling processes with applications around 0 °C, such as in the food industry, ice storage units are often used, and in industrial refrigeration networks storage units of -20 °C are common.
- **Location**:
  - The location of the energy storage is an important criterion for characterizing an energy storage system. A distinction is made between centralized and decentralized energy storage systems. As a rule, a heat storage facility set up at the location of heat generation is referred to as centralized, whereas heat storage facilities at the heat consumers are referred to as decentralized.
  - Most thermal energy storage units are stationary, whether central or decentralised systems. However, there are also mobile heat accumulators that do not transport heat to a heat consumer. The density of the energy storage plays a major role here, and latent heat storage or thermochemical storage systems are usually used on a mobile basis, although only in a few pilot plants.
- **Storage capacity and thermal performance**:
  - High storage capacity and energy density are important when space is limited or large amounts of energy need to be stored.
  - Storage tanks with high thermal performance are used in areas such as hot water supply and district heating. This is possible with sensitive hot water storage tanks in which the required hot water is also the storage medium.
- **Duration of storage**: An essential feature of all energy storage units is the duration of the period to be bridged between charging and discharging. Short-term storage tanks are used for periods ranging from hours to a few days, while long-term storage tanks can store energy for weeks to a year. As a rule, long-term storage units must be able to store large quantities of thermal energy at relatively low charging and discharging rates, while short-term storage units have high thermal capacities and lower stored heat quantities.
- **Storage capacity**: Storage capacity is the amount of energy that a storage unit can make available to the consumer under certain process conditions. From an exergy point of view, the storage capacity depends decisively on the ambient temperature. If the storage capacity is assessed technically, the actual temperature change in the storage tank is of importance (see chapter 2.10.3.1). If only one storage medium is heated, the capacity is directly proportional to the temperature difference. If a phase change takes place when the temperature of the medium increases, the melting or evaporation enthalpy also contributes to the increase in capacity. The storage capacity is given in absolute terms in J or kWh and as a specific quantity, for example volumetrically in kWh/m3. Economically, the storage capacity must be co-related to the costs of the storage facility.
- **Charge and discharge capacity**: The charge and discharge capacity describe how quickly a storage unit can absorb and release a certain amount of heat. It is decisive for optimising a storage unit to suit the charging and discharging profile specified by the application. In contrast to capacity, performance does not depend on the physical storage effect but on the mass flow or heat transfer and temperature difference. The dimensioning of the incoming and outgoing cross-sections or the technical design of, for example, a heat exchanger or reactor with specific transfer surfaces, is of importance. The capacity of a storage tank is often specified as nominal capacity under design conditions and depend on current process parameters. The capacity can be expressed as an absolute value in W or also on the volume or mass of the storage tank in W/m3 or W/kg. For an economic assessment of a storage facility, capacity and costs are co-related. An equally common representation, especially for latent heat storage tanks, is the relation of capacity (in kWh) relative to performance (in kW). This quotient results in the time needed for charging and discharging and can be different for either.
- **Efficiency**: Efficiency describes the ratio of the usable energy during discharging to the energy absorbed by the storage unit during charging. The useful heat is less than the supplied heat because of losses in

the storage tank. Likewise, the stored exergy decreases with decreasing temperature. The efficiency is dimensionless and is usually given in percent. See also chapter 2.6.1 for the term efficiency.

- **Storage cycles**: The period between the loading and unloading process is called storage duration (also called storage period) (Figure 2.29). The sum of the charging, storage and discharging times (and possibly also a standstill period in the discharged state) forms a storage cycle. What storage cycles look like depends on the time profiles of the heat source and the consumers. The storage capacity and the possible charging and discharging capacity also play a role.

With thermal energy storage systems, these times range from minutes to a year. If irreversible processes take place within a storage cycle that affect the storage capacity, the number of executable storage cycles is limited. The stability of thermal energy storeage is determined by their degradation over a certain number of cycles. If a storage system is operated with many cycles per year, this has a positive influence on the cost effectiveness. The system then converts energy, provides stored energy and is thus amortized faster.

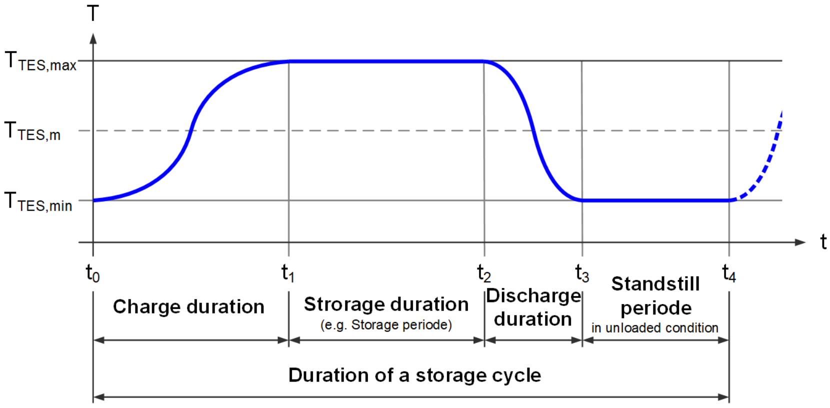


Figure 2.29 Storage tank temperature as a function of time and terms used to describe a storage cycle.

2.10.2 Storage technologies

Thermal energy can be divided into different storage technologies. For the application in district heating networks mainly the sensitive heat storage is used, as this technology is the most developed and relatively inexpensive. For the sake of completeness, latent and thermochemical heat storage are also introduced, but these are practically irrelevant for use in district heating networks.

2.10.2.1 Sensible thermal energy storage

In sensible heat storage, a storage medium is heated or cooled without phase change. The amount of stored energy depends on the specific heat capacity of the substance in kJ/(kg K). In most cases, water is used because of its high specific heat capacity and it is also environmentally friendly, cost-effective, thermally stable and available practically everywhere.

Unpressurized storage tanks with water is used for a temperature range 0 °C to 100 °C. Above 100 °C storage tanks with water are under pressure. Typical temperatures are up to 200 °C for process heat. Above 100 °C the vapour pressure rises sharply, which places high pressure demands on the storage tanks and leads to correspondingly higher costs.

Thermal oil can be operated at higher temperatures of up to 400 °C depending on the type. The thermal conductivity of thermal oil is considerably lower than that of water and the heat capacity is about half that of water. By combining thermal oil with a rock stratification of high heat capacity, the storage capacity can be increased and the amount of relatively expensive thermal oil reduced. This combination is comparable to a gravel/water storage tank for low temperature applications. For higher temperatures, solid storage materials such as concrete or special ceramics are also suitable.

For temperatures below 0 °C water-glycol mixtures are used.

Depending on the temperature stroke and the application-specific loading and unloading temperatures, latent and thermochemical heat storage systems generally have higher energy densities than sensible heat storage systems. For the sensible application of latent heat storage tanks, the maximum temperature rise is around 20 K (e.g. ice storage tanks). At a temperature rise of more than 50 K, a sensible heat accumulator with water shows higher energy density. They have been around longer are considerably cheaper.

In the case of sensible energy storage systems, the temperature difference between the storage medium and the environment is usually relatively large, so thermal insulation plays an important role in reducing the relatively high self-discharge during the storage period.

2.10.2.2 Latent heat storage

With latent heat storage systems, the energy required for a phase change is stored in addition to the sensible heat. In practice, the solid-liquid phase transition is usually used. The volume change is generally less than 10 % and is technically controllable. At the same time the phase change enthalpy is sufficiently high.

Latent heat accumulators have several advantages over sensible heat accumulators. For example, considerably more thermal energy can be stored at small temperature differences. For such applications they have a higher energy density and can be built much more compactly due to the high storage capacity. In addition, the temperature during charging and discharging is constant over a long period of time, which enables the use of Phase Change Materials (PCM) integrated into the building structure, for example.

Phase Change Slurries (PCS), also known as slurries, are suitable for the transport of thermal energy. Slurries have the advantage that they are always pumpable, regardless of the state of the aggregate. This is used, for example, in the air conditioning of buildings or in industrial plants.

The disadvantages are the relatively high costs and the fact that the current state of development of the technology is not yet suitable for use in district heating.

2.10.2.3 Thermochemical heat storage

The term thermochemical energy storage refers to chemically reversible reactions in which the reaction products can be separated and stored for a long time. The separation does not cause any storage losses and only when discharged does the exothermic reaction release the stored energy again. These systems allow very high energy densities, but are currently hardly used, as the technology is still in the basic research stage.

2.10.3 Fundamentals of energy storage

2.10.3.1 Storage process

Thermal energy storage systems increase their energy content (charging) by supplying energy or an energy carrier, store this energy content over a certain period of time with as little loss as possible (storing) and release energy or the energy carrier again when required (discharging). When discharging, the energy content of the storage unit is reduced (Figure 2.30).

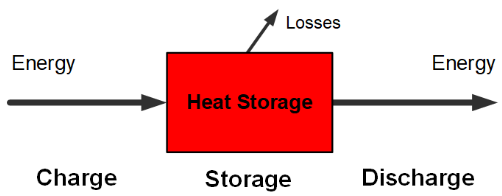


Figure 2.30 Process of the thermal energy storage. Load – Storage - Unload

2.10.3.2 Thermal energy

Actually, the term energy storage is not correct from a physical point of view, because energy is not consumed but can only be converted and in this sense is not stored. Rather, the storage unit has the thermodynamic potential to transfer a certain amount of heat or cold to another medium. Consequently, the term **exergy** describes the proportion of energy that can be converted into other forms of energy without restriction under ambient conditions. The higher the difference between storage and ambient temperature, the greater the exergy content and the greater the value of the heat. The term **anergy**, on the other hand, describes the proportion of energy that is released into the environment and is therefore no longer usable. When considering a lossy loading, storage and discharging process of a thermal energy storage device, the anergy content increases and the exergy content decreases (Figure 2.31). The sum of the anergy and exergy content remains constant.

The following equation shows the calculation of the exergy content.

$$E_{ex} = Q \cdot \frac{T_{TES} - T_{AMB}}{T_{TES}} = Q \cdot \left(1 - \frac{T_{AMB}}{T_{TES}}\right)$$

E_{ex} = Exergy content at the storage device [kWh]
Q = Energy content at the storage device [kWh]
T_{TES} = Storage temperature [K]
T_{AMB} = Ambient temperature [K]

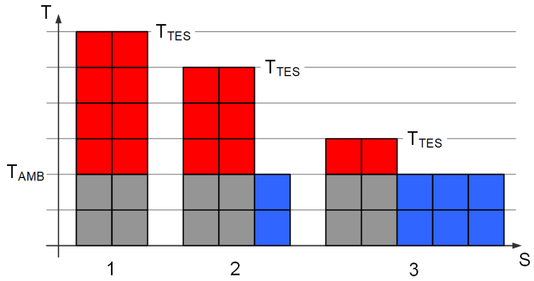


Figure 2.31 Energy content in the heat accumulator (T,s-diagram according to [34]). The exergy content (red) in the storage tank is converted into energy (blue) during storage and discharge. The energy as the sum of the two terms remains constant.
1) Storage tank fully charged
2) Storage included heat losses
3) Heat release (discharge).

It can be seen that the exergy content depends on the heat quantity Q, the storage temperature T_{TES} and the ambient temperature T_{AMB}. The fractional term in brackets $(1\text{-}T_{low}/T_{SpLY})$ refers to the Carnot-Efficiency i.e. the maximum working capacity contained in the heat.

For district heating applications, only sensible heat storage is an option. If the energy content in the storage tank or the amount of heat contained therein is examined, sensible heat is understood to be the heat absorption or release that results in a noticeable change in temperature. The storage medium used is heated or cooled. The relationship between the heat quantity Q contained in the storage medium and the temperature change T is described as follows:

$$Q = m\, c_p\, \Delta T$$

Q = Heat Quantity [kWh]
m = Quantity of the thermal energy storage medium within the storage device [kg]
c_p = specific heat capacity of the thermal energy storage medium [J/ (kg K)]
ΔT = Temperature variation within the storage device [K]

In addition to the temperature change T and the material mass m, the heat quantity Q depends on the specific heat capacity c_p of the storage medium. A high specific heat capacity is helpful for heat storage, since a smaller quantity of the storage medium is required and the storage unit is therefore smaller. Of all storage media that can be used in practice, water has the highest specific heat capacity per mass at around 4.2 kJ/(kg K). In terms of volume, water also has the highest value, i.e. higher than metals or natural materials such as stone. A gravel bed storage tank thus has a lower heat capacity in total than a pure water storage tank.

2.10.3.3 Thermal insulation

Thermal insulation plays a particularly important role in reducing thermal losses in sensible heat storage systems. The connection between temperature differences and heat losses of a storage tank is described as follows:

$$\dot{Q}_L = U\,A\,(T_{TES} - T_{AMB})$$

$\dot{Q}_L$	= Heat loss storage device [W]
U	= Heat transfer coefficient [W/ (m^2 K)]
A	= Surface storage device [m^2]
T_{TES}	= Storage temperature [°C]
T_{AMB}	= Ambient temperature [°C]

Along with the surface area A of the storage tank, the storage tank temperature TTES and the ambient temperature T_{AMB}, the heat transfer coefficient U is the decisive characteristic value of thermal insulation. The U-value is given in W/(m^2 K) and describes the power per Kelvin temperature difference transmitted through one square meter of surface. Depending on the design of the storage tank (sphere, cylinder, etc.) the heat transfer coefficient must be calculated individually. For a flat wall with n-layers it can be determined as follows:

$$U = \frac{1}{\frac{1}{\alpha_i} + \frac{s_1}{\lambda_1} + \frac{s_2}{\lambda_2} + \ldots + \frac{s_n}{\lambda_n} + \frac{1}{\alpha_a}}$$

U	= Heat transfer coefficient [W/ (m^2 K)]
s	= Insulation thickness [m]
α	= Heat transfer coefficient [W/ (m^2 K]
λ	= Heat conductivity [W/ (m K]
i	= Inner (Storage Device)
A	= Outer (Ambient)

The heat transfer coefficient thus depends on the thermal conductivity and layer thicknesses s_i of the insulation materials as well as on the heat transfer coefficients on the inside and outside. The proportionality factor is α which describes the intensity of heat transfer at an interface. The heat transfer coefficient is not a material constant, but depends on the flow velocity, the type of fluid, the geometric conditions and the surface quality.

With common **insulation materials** such as porous foams or fibre insulation materials, air convection is suppressed by a fine-porous structure. The insulation material itself must be a poor heat conductor. The insulation materials are offered either as panels or as pourable powder, which allows them to adapt optimally to any storage geometry. Flexible non-woven insulating materials such as mineral or glass wool are also common, but have the disadvantageous tendency to absorb and store moisture. Commercially available insulation materials have a thermal conductivity of 0.025 to 0.07 W/(m K).

In addition to the choice of insulation material, the following aspects must also be considered to achieve good thermal insulation:

- Water vapour diffusion (or small leaks) lead to damp insulation layers, which therefore insulate worse.
- The insulation material can age due to solar radiation and temperature stress.
- In the case of panel materials, irreversible air gaps and thus convection and chimney effects can occur due to negligent installation or thermal expansion during operation.
- In the case of powder spills, the material can settle so that an uninsulated air-filled space with high heat loss is created, especially in the upper area, which usually has the highest temperature.
- The insulation of the lower storage tank area should not be neglected, as large losses can temporarily occur there at higher return temperatures and when the storage tank is fully charged.
- Due to many or poorly insulated connections and the necessity of a support frame for installation in the floor area, losses of up to half of the total heat loss of the wall insulation can occur.
- Finally, natural convection in the supply lines (single-pipe circulation) must be excluded, preferably by means of thermosyphons or by installing convection brakes.

A particularly high insulating effect can be achieved by a vacuum (similar to the double-walled thermos flask), which replaces the insulating material. However, due to the size of the heat storage tanks used in the district heating sector, this type of insulation is generally not used.

2.10.3.4 Cooling by time

The cooling of a sensible storage system is described with the exponential function by the following equation:

$$\Delta T(t) = \Delta T_0 \cdot e^{-\frac{A\,U}{V\,\rho\,c_p}t}$$

$\Delta T(t)$	= Temperature difference for time set point t [K]
ΔT_0	= Temperature difference at the initial set point [K]
A	= Surface area storage device [m^2]
V	= Volume storage device [m^3]
U	= Heat transfer coefficient [W/ (m^2 K)]
ρ	= Density thermal energy storage medium [kg/m^3]
c_p	= specific heat capacity storage medium [J/ (kg K)]
t	= Observation time [s]

Where ΔT(t) is the temperature difference between the storage temperature and the ambient temperature at time t and ΔT_0 is the temperature difference between the storage temperature and the ambient temperature at the starting time. It can be seen that for the longest possible cooling times during storage not only the heat transfer coefficient but also the surface-to-volume ratio (A/V) of the storage as well as the density and the specific heat capacity c_p of the heat storage medium play an important role.

Figure 2.32 shows an example of the cooling of a storage tank for a heat load demand of 1,000 kW. With a fully charged storage tank and a temperature difference of 30 K, the heat load demand can be covered for one hour if the storage tank has a volume of 29 m^3. Based on a cylindrical storage tank with a height-diameter ratio of almost 7.3 (see also example on next page: diameter 1.1 m, height 8 m) the surface-to-volume ratio A/V is 2.10. It is assumed that the storage tank is fully charged at a temperature of 80 °C and the average ambient temperature is 10 °C. The average heat transfer coefficient of the storage tank shell is 0.2 W/(m^2 K).

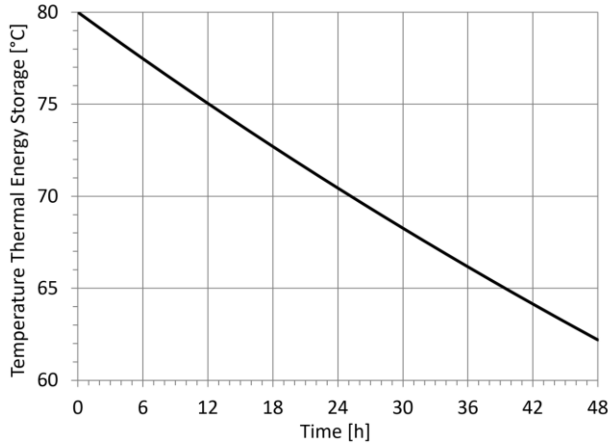


Figure 2.32 Cooling down the storage tank temperature as a function of time. Storage tank dimensioning for a load demand of 1'000 kW, compensation for one hour with a temperature difference of 30 K. Storage temperature from 80 °C and an ambient temperature of 10 °C

2.10.4 Storage tank design and dimensioning

Thermal energy storage systems can be realised in a variety of different designs. The designs result from the storage technology used and the requirements of the respective application. For the sensible heat storage used in the district heating sector, the so-called direct loading has proven to be the simplest design. Here the heat storage medium is also the heat transfer medium, so there is no hydraulic separation. From a chemical point of view, the storage unit contains only one component that changes its temperature, but not its aggregate state. The advantage of such direct charging is the simple design and, with the exception installations for optimum inflow, the absence of internal fittings. Thus, the space is filled only by active storage material. With indirect charging, heat is transferred to the storage medium via heat exchangers.

This concept is used when different fluids are used in the heat supply (e.g. water-glycol/water) and/or different system pressures exist between the network and the storage tank. An advantage is the protection against corrosion, contamination and loss of storage material, since the storage tanks, as closed systems, have no material connection to the environment. A disadvantage is the relatively high exergy loss via the heat exchangers.

According to the basic equation of heat output, the charging and discharging capacity of a directly or indirectly charged storage unit depends not only on the temperature spread but also mainly on the maximum mass flow of the storage medium:

$$\dot{Q} = \dot{m}\, c_p\, \Delta T = \dot{V}\, \rho\, c_p\, \Delta T = v\, A\, \rho\, c_p\, \Delta T$$

The maximum flow velocity in the reservoir thus has a direct influence on the diameter of a cylindrical reservoir (Figure 2.33). As a guideline value to obtain the temperature stratification in the storage tank, the usable charging and discharging capacity is limited by a maximum flow velocity in the storage tank of 6 m/h (0.0017 m/s) to 10 m/h (0.0028 m/s).

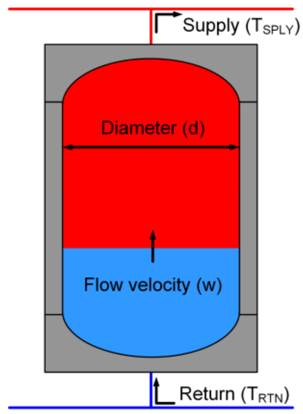


Figure 2.33 Display of flow velocity and diameter of a directly loaded energy storage device with ideal temperature stratification (red = high temperature, blue = low temperature)

Depending on the dimensioning and available temperature difference between flow and return flow, high charging and discharging capacities can be achieved. Applying the basic equation of heat output, the following relationship with $A = d^2\, \pi/4$ results for the diameter in a pipe or a cylindrical tank:

$$d = \sqrt{\frac{\dot{Q}\; 4}{w\; \rho\; c_p\; \Delta T\; \pi}} = 0.0176\,\sqrt{\frac{\dot{Q}}{w\; \Delta T}}$$

d = Diameter within the storage device (m)

$\dot{Q}$ = Load- or discharge capacity of the storage device (kW)

w = Flow velocity within the storage device in (m/s); 0.0017 m/s (6 m/h) until 0.0028 m/s (10 m/h)

ρ = Water density (kg/m^3)
Simplification for water of 60 °C (983 kg/m^3)

c_p = Heat capacity of water in (kJ/ (kg K))
Simplification for water of 60 °C (4.183 kJ/ (kg K))

ΔT = Temperature difference between Supply- and Return (K)

Calculation example: For a charging and discharging capacity of 1,000 kW, a maximum flow velocity in the storage tank of 0.0022 m/s (8.0 m/h) and a temperature difference of 30 K, the diameter of the storage tank is approximately 2.2 m (Figure 2.34). To provide this charging and discharging capacity for one hour, a cylindrical storage tank must be at least 8 metres high. This results in a volume of around 29 m^3.

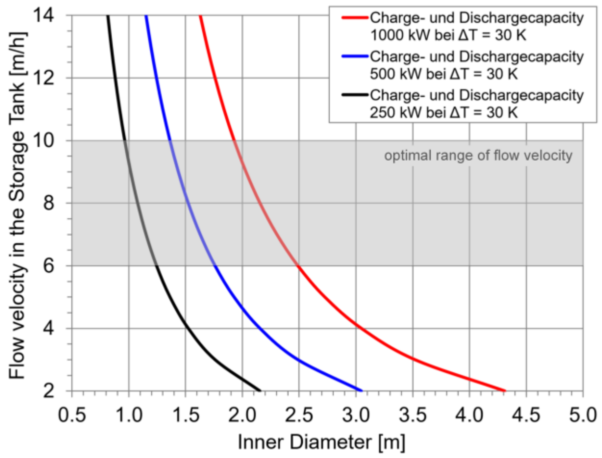


Figure 2.34 Comparison of the diameters of the storage tank as a function of the flow velocity in the tank according to the above equation for a charge and discharge capacity of 250 kW, 500 kW and 1'000 kW at a temperature difference of 30 K.

In order to make optimum use of heat storage, temperature equalisation between hot and cold zones in storage units is necessary. In water storage tanks, the **temperature stratification** caused by density differences must be maintained, as shown in Figure 2.33, with a hot layer in the upper zone and a cold layer in the lower part of the tank. The design of the **inflow** or **outflow** of a storage tank is particularly decisive for this. The impulse generated by the inflowing mass flow causes turbulence in the reservoir, which can destroy the thermal stratification. The strength of the turbulence created by the inflow depends on the type of inflow, the formation of thermal stratification and the inflow impulse.

Preferred **designs** of inlet and outlet are, according to Figure 2.35, e.g. nozzle pipes or circular deflector plates with horizontal flow outlet (also called radial diffusers). However, directly connected pipes, pipe elbows near the floor or with deflector plates have proved to be unfavourable. To calm the inflow in unfavourable designs, installations such as perforated plates and/or funnel-shaped outlets (diffuser) should be provided. In general, connections and fittings of all kinds should be dimensioned sufficiently large to keep the inflow velocity and thus the formation of vortices low. An inflow velocity of 0.1 m/s to 0.2 m/s is recommended as a practical value.

The **height-diameter ratio** is another key figure that results from the dimensioning and design of the storage tank. There are two opposing points to consider. On the one hand, an optimum surface-to-volume ratio with minimal heat losses is the aim. On the other hand, there are guide values for the optimal formation of temperature stratification in the storage tank, e.g. for recording the storage tank state of charge.

Figure 2.36 shows the cooling of the storage tank for different storage tank sizes or surface-to-volume ratios. The dimensioning of the storage tank was created for a thermal heat load demand of 1'000 kW. The storage tank volumes were determined for compensation of the required heat for 1, 12 and 24 hours at a temperature difference of 30 K. The storage tanks have a volume of 29 m3, 350 m^3 and 700 m^3 and a surface-to-volume ratio of 1.8, 0.79 and 0.62 m^2/m^3 respectively. The surface-to-volume ratio is based on a cylindrical heat storage unit with a height-to-diameter ratio of 1, resulting in an optimum surface-to-volume ratio that comes closest to that of a sphere.

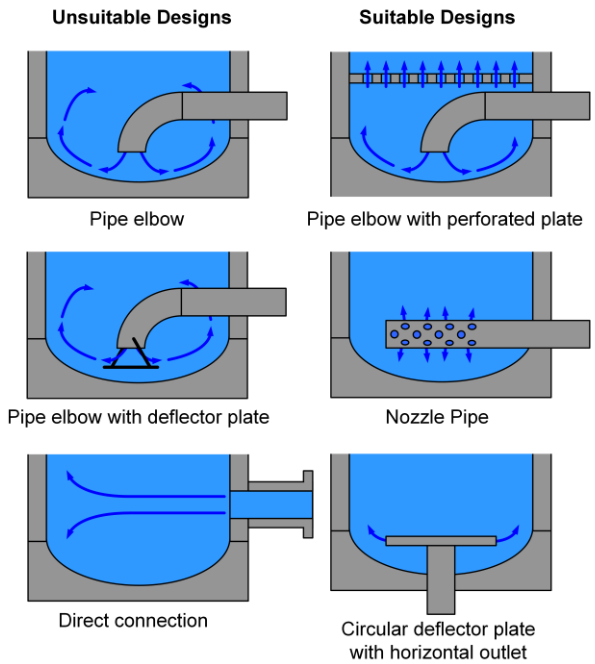


Figure 2.35 Suitable and unsuitable designs of the inflow and outflow of a storage tank.

It is further assumed that the storage tank is fully charged at a temperature of 80 °C and the average ambient temperature is 10 °C. The average heat transfer coefficient of the storage tank wall is 0.2 W/(m^2 K) for all three tanks. For comparison, the storage tank shown in Figure 2.32 has a surface-to-volume ratio of 2.10 m^2/m^3 and a volume of 29 m^3.

It is clearly visible that the surface-to-volume ratio or the height-to-diameter ratio has a considerable influence on the cooling behaviour. On the other hand, a height-diameter ratio of 5 to 10 is recommended for the optimal formation of a temperature stratification. This is illustrated by the red dotted line in Figure 2.36. This corresponds to the same attributes as the red extended line, but with a height-diameter ratio of 7.3 instead of 1. With a thicker insulation layer, the higher heat losses can be compensated for with larger height-diameter ratios. However, this is usually only profitable for smaller storage tanks.

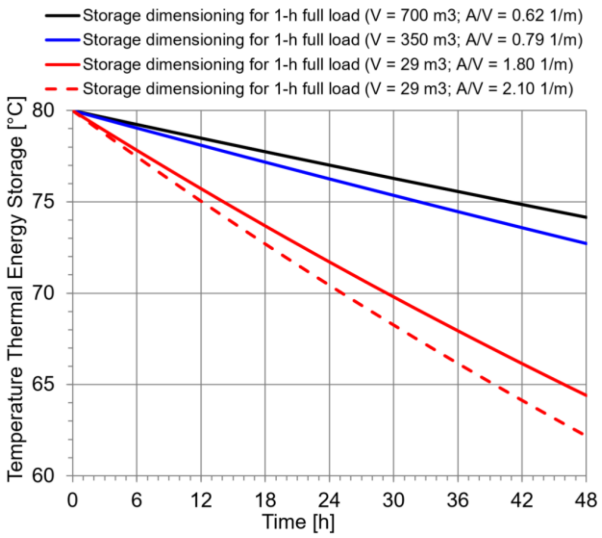


Figure 2.36 Cooling of the storage tank as a function of time. Storage tank dimensioning for a thermal heat load demand of 1'000 kW, compensation for 1, 12 and 24 hours under full load and a height-diameter ratio of 1. The dotted line corresponds to a height-diameter ratio of 7.3. Initial storage tank temperature of 80 °C and an average ambient temperature of 10 °C. 1/m = m^2/m^3.

Depending on the application, space requirements and the necessary volume of the storage, several storage units may be necessary. A distinction must be made between serial and parallel operation. Serial operation of the units is only recommended if the maximum flow velocity in the storage unit of 6 m/h to 10 m/h is not exceeded (see charging and discharging capacity). Otherwise, parallel operation is recommended. During parallel operation the principle of constant pressure drop must be observed (Tichelmann principle). The principle is that the medium flows the same distance everywhere (same pipe diameter, same length). As a result, the same pressure losses occur over each element, so that all parts are heated equally or are filled or discharged evenly, as is the case with storage tanks connected in parallel, for example. It is also possible to manage several storage tanks via dampers either in serial or parallel operation. This can be useful, for example, if the effective charging and discharging capacity cannot be conclusively determined. However, this is more complex and costly.

Storage tank sizes up to 4.5 m in diameter are suitable for road **transport**. Larger heat storages (diameter) must be welded or concreted on site and are usually no longer pressure-resistant. With a diameter of 4.5 m and a height-diameter ratio of 5 (height 22.5 m), this results in a volume of almost 358 m^3.

The **compressive strength** of the storage tank should only be set as high as necessary. Since a storage unit is usually integrated in the return flow, it adapts to the lower return pressure. Commercially available reservoirs are usually offered for a pressure resistance of 6 bar to 10 bar. Over 15 bar the necessary wall thicknesses are too large to be cost efficient. Storages up to 100 °C are designed pressure-less with gas cushioning (e.g. with nitrogen). Storages above 100 °C (pressure accumulators) are designed without pressure pads.

Long-term storage tanks for heating and cooling applications in the temperature range from 0 °C to 100 °C are called underground heat storage tanks:

- Aquifer storage tank
- Geothermal probe storage
- Cavern storage
- Gravel/water or soil/water storage.

Aquifer and geothermal probe reservoirs have the disadvantage that they cannot be thermally insulated. They therefore have greater heat losses than insulated hot water tanks and gravel/water tanks. It should also be noted that they require several years to reach a steady state in order to operate economically. Therefore, the efficiency is not very high at the beginning. This is due to the fact that at the beginning the storage volume and the surrounding soil must be heated to operating temperature. In the area of long-term storage, such as for the seasonal storage of solar heat or in the case of CHP plants two topics are in the foreground: innovative insulation concepts, which must be characterized above all by cost reduction, and suitable stratified-charge units, which store the heat transfer fluid in the appropriate temperature layer when loaded directly.

2.10.5 Integration and application

Thermal energy storage systems play an important role in the heat supply. Thermal energy storage systems in district heating can be applied solely in the heat sector or also combined in the electricity and heat sector.

2.10.5.1 Integration in the heat sector

The application of thermal energy storage in the heat sector and especially in the district heating sector can be very versatile. The focus here is on a heating plant where heat is generated centrally and then distributed on site or via a heat distribution network. Here, the main task of the thermal energy storage system is to compensate for power peaks as well as power reductions so that the heat generation plant has enough time to react to changes in demand. This is particularly necessary for energy generators with high inertia, such as wood-burning systems. Depending on the fuel water content, automatic wood firing systems achieve a change in output of approximately one percent per minute. This means that a wood firing system needs about 70 minutes from the regular operation of 30% output to the full heat output of 100%. With a thermal energy storage system, the operating behaviour of the wood boiler can be significantly improved by compensating load peaks and load reductions through the heat storage system, while the wood boiler can follow the average heat load demand. This means that the wood boiler can be dimensioned smaller, runs longer and more efficiently and causes lower pollutant emissions. According to QM Holzheizwerke [21], a storage capacity to compensate the nominal heat output of the wood boiler for one hour at a temperature difference of 30 K is recommended. In multi-boiler systems, the storage capacity should be individually adapted to the demand, but should at least correspond to the nominal heat output of the larger wood boiler. If the temperature difference over the storage tank is less than 30 K, the storage volume should be adjusted accordingly.

Special attention should be paid to the operation of the storage tank. When managing a thermal energy storage unit, it is important, as described in chapter 2.10.3, that the temperature stratification in the storage unit is not disturbed. This can be ensured with suitable installations and generously dimensioned inlet and outlet cross-sections as well as hydraulic balancing. In the case of heat generators with a sluggish operating behaviour, such as wood boilers, attention must also be paid to the recording of the storage tank charge status and the resulting output specification for the wood boilers. In this case, QM Holzheizwerke [22] makes recommendations that can also be applied to other systems.

Another possible application of storage tanks is thermal networking (anergy networks or LowEx application) of different heat sources and heat sinks at a relatively low temperature with geothermal probe or ground collector field storage tanks for seasonal storage. Central or decentralised heat pumps can be used. Large seasonal heat storage tanks are used in solar thermal energy in Germany and Denmark, for example [36], [37] and [38]. In this case, the main task of the energy storage unit is to compensate for any asynchronous heat generation and heat demand. Depending on the application, the storage period can be days, several weeks or up to one year.

In addition to conventional storage charge management such as maintaining a constant state at 60 %, weather conditions can also influence the storage charge management. This is particularly important where the weather has a considerable influence on the heat demand, such as in greenhouses. By taking the weather forecast into account, the climatic influence can be determined up to 7 days in advance. If the storage tank is dimensioned sufficiently large and has a capacity of 4 to 8 hours of the nominal heat generation capacity at a given temperature difference, the heat generation can be dimensioned much smaller and operated at a significantly higher capacity than without a storage tank. The weather forecast can an important factor in e.g. the thermal cross-linking of different energy sources, especially solar thermal energy.

Depending on the system, the use of thermal energy storage systems thus offers considerable potential for optimising plant operation in terms of costs and primary energy consumption. The most important points are:

- - Thermal energy storage systems reduce peak heat demand
- - Heat generation plants can be designed smaller.
- - Heat storage systems have a positive influence on the operating behaviour of the heat generation plant
- - Heat storage systems form an energy reserve for faults or short interruptions
- - Waste heat can also be used if it is not generated at the same time as the heat demand
- - Decentralised heat storage systems can relieve the network at neuralgic points in the district heating network and enable the connection of further consumers in a busy network.

2.10.5.2 Integration of a combined Heat and Power Unit

For CHP plants, a distinction is made between heat-controlled and current-conducted operation. In Switzerland, decentralised CHP plants are generally heat operated, as the electric efficiencies are low and the release of waste heat into the environment is restricted by the requirements of cost-covering feed-in tariffs as in Figure 2.9. In heat-driven operation, the CHP plant operates according to the heat load, while electricity is generated as a by-product. However, this has the disadvantage that electricity is generated at times when it is not needed in the grid. Heat-operated CHP plants can thus increase the storage demand in the electricity sector. This can be avoided by partially decoupling the production of heat and electricity in CHP plants through the use of heat storage and other measures. Flexible applications of CHP plants can optimise plant operation in terms of costs and primary energy consumption by including heat storage, other components such as peak load boilers and heat pumps, a steam turbine bypass [39] or the use of decentralised electricity storage. However, bypassing the electricity production is energetically undesirable, but electricity storage is considerably more expensive than heat storage. If no fixed feed-in tariff is paid for electricity production, the use of heat storage units in CHP plants is therefore becoming increasingly interesting. Co-generation plants can thus be used at least in part for the demand-based production of expensive control energy (power-on-demand), while the district heating network is partly supplied by heat from heat storage. At the same time, heat storage facilities offer potential savings in the dimensioning of the plant output.

3 Connection heat supply and heat distribution

In this chapter, the most important principles and components for the supply and distribution of heat are presented.

The pump unit with the associated control equipment and the expansion and pressure control system are the link between the heat supply and heat distribution. For certain requirements it may be necessary to separate a network or to use a pressure boosting station.

3.1 Network Temperatures

The supply and return temperatures of the network result from the demands of the heat consumers for a certain supply temperature, the greatest possible temperature difference or for the lowest possible return temperature.

3.1.1 Supply temperature

The demands of the heat consumers are very different (Table 3.1) and the supply temperature of the district heating network must always be based on the highest necessary supply temperature. Depending on the situation, the heat consumers can be grouped in different temperature levels (e.g. high temperature for radiators and low temperature for floor heating systems). In this case a cascading of the supply temperature can be advantageous, in that the return flow of a heat consumer with high temperature serves as supply for downstream heat consumers with lower supply temperatures.

Table 3.1 Required primary supply temperature for different heat consumers.

Heat Consumer	Supply-Temperature
Process Heat	
Hospital with sterile steam generation 3 bar	≥ 160 °C
Drying process from the food technology	≥ 130 °C
Industrial plant with secondary domestic hot water network 80/60	≥ 85 °C
Spa and wellness resort (Hygiene Circuit)	≥ 70 °C
Greenhouse with air heating	≥ 60 °C
Greenhouses with floor heating	≥ 40 °C
Space Heating and Domestic Hot Water (DHW)	
Building with radiator (with or without DHW)	≥ 65 °C
Building with low temperature heating (without DHW)	≥ 40 °C

In the case of **extensive networks**, the temperature difference has a strong influence on investment and operating costs. If the temperature difference is halved, the required volume flow is doubled for the same capacity, so that the energy required for pumping capacity increases accordingly. However, a design for a pipe diameter larger by a factor of about 1.3 is required, which increases the investment costs. As a result, the heat losses of the network also increase, which further raises the heat generation costs.

For the cost efficiency of networks, the temperature difference and thus the supply temperature must increase with increasing length of the network. Higher supply temperatures lead to higher heat losses.

The **conditions of the heat source** have the following influences on the supply temperature:

- The supply temperature is given by the temperature of the available waste heat.
- Limitation of the supply temperature when extracting steam from a process at a defined steam pressure. A high extraction temperature can therefore only be guaranteed by reducing the power generation.
- A heat extraction at the source already exists at a defined temperature level (e.g. geothermal energy, heat pump, CHP system).

The **choice of the supply temperature** has a technical influence on the choice of pressure level, the dimensioning of the pipe cross-sections, the pump delivery rate and the heat losses. A high supply temperature may allow a lower pressure rating under certain circumstances or otherwise smaller pipe cross-sections and lower pump efficiency. At the same time, heat losses increase and, depending on the type of heat generator, its energy consumption increases. This is especially true for combined heat and power and heat pumps. For this reason, the supply temperature and pipe cross-section must be optimised.

3.1.2 Return Temperature

The return temperatures result from conditions such as age and the building structure to be heated, type of hot water preparation, type of hydraulic integration at the heat consumers. There are also requirements for the return temperatures from the heat source. In certain cases, heat utilisation requires low return temperatures. This applies, for example, to exhaust gas condensation systems.

Table 3.2 can be used to obtain practical values and target values for return temperatures.

The maximum return temperatures should be defined in the heat supply contract or in the technical connection regulations TCR (see chapter 5.3). A differentiated requirement of the maximum return temperature for the heating case and for domestic hot water heating is recommended.

In order to generate the lowest possible return temperatures, an energy price that depends on the return temperature could create the incentive for this. For existing and also new district heating networks, the return temperature or the temperature difference should be evaluated. This evaluation will quickly identify optimisation potential and malfunctions in transfer stations. A simple method is described in detail in chapter 10.1.2, with the recording of additional consumption.

Table 3.2 Standard - and practice-oriented values for return temperatures.

Heat Network		Return-Temperature
Waste incineration plant > 10 MW	Practice-oriented Value	55-65 °C
Wood-fired furnace with flue gas condensation	Practice-oriented Value	> 45 °C
	Target Value	≤ 45 °C
Housing estate	Practice-oriented Value	> 38 °C
	Target Value	≤ 35 °C

3.2 Network operation

Supply temperature adjustment and progression should be recorded in the technical connection regulations (TCR) of the respective heat supplier. There are three different operation modes to ensure the network supply temperature (Figure 3.1).

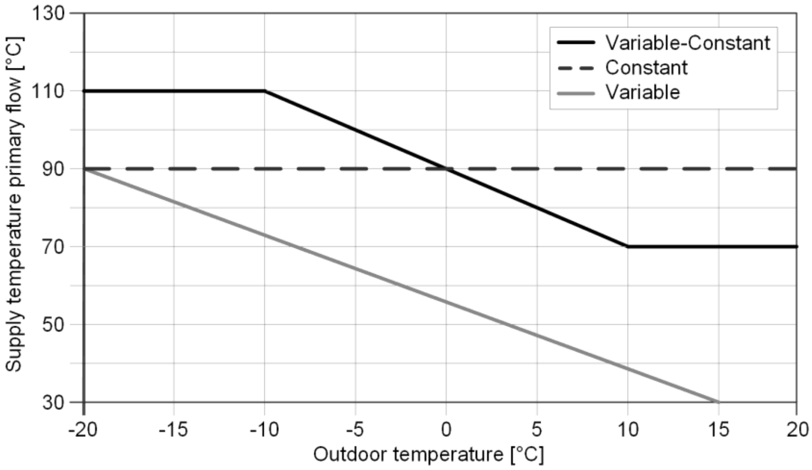


Figure 3.1 Supply temperature from the heating plant as a function of the outside temperature for the three different operation modes of district heating networks (variable-constant, constant, variable).

3.2.1 Variable operation mode

The network supply temperature is controlled depending on the outside temperature. When the outdoor temperature falls, the network supply temperature is raised gradually up to the maximum value. If the outside temperature rises, the network supply temperature is reduced gradually until the heating limit is reached and the heat supply is stopped. The variable mode is only suitable for supplying room heating. It is not suitable for the use of process heat and hot water which are independent of the weather.

3.2.2 Variable-constant operation mode

The network supply temperature is controlled within defined limits depending on the weather conditions. If the outdoor temperature falls, the network supply temperature is raised gradually up to the maximum value. If the outdoor temperature rises, it is reduced to the minimum value. The minimum value is determined by the minimum network supply temperature to be maintained (e.g. hot water heating). The variable-constant mode is the most common and allows the simultaneous supply of room heat, hot water and process heat. By readjustment to the secondary supply temperature required by the consumer, an operating mode independent of the heat supplier is possible with regard to supply temperature and heating time.

The current outdoor temperature is rarely used as the command variable for the network supply temperature. Depending on the size of the network, it may be advisable to use the outdoor temperature averaged over a longer period of time or a temperature forecast.

3.2.3 Constant operation mode

The network supply temperature is kept constant regardless of the outside temperature. In principle, all common heat consumers can be connected if the constant supply temperature offered is sufficient for the intended use. A supply temperature control according to the requirements of the respective heat consumer must be provided locally. Due to the constant operation, it is possible to offer the heat output even at higher outside temperatures, which is particularly useful for process heat and hot water heating. However, this operation mode leads to increased heat distribution losses in the transition period and in summer operation.

3.3 Pumps

Circulating pumps driven by an electric motor are used for water circulation in district heating networks. These are either operated at constant speed or - as is prescribed today - are infinitely variable. Figure 3.2 shows a typological overview of the designs and applications of the different pump types as a function of the delivery head and volume flow. In addition to the distinction between axial and radial design, a distinction is also made between dry-running and wet-running pumps (glandless pump).

3.3.1 Designs

Dry-running pumps are used for larger district heating networks. These are connected to a standard motor via shaft and coupling, whereby a distinction is made between the types of base pump (motor and pump mounted on a base) and in-line pump (motor mounted on a built-in pipe pump). Table 3.3 shows the typical delivery heads and volume flows of the pump types, whereby the values apply to a single pump. Depending on the configuration in series or parallel connection, higher values can be achieved, as shown in Figure 3.2.

In building systems and also in small district heating networks, **wet-running pumps** are used with a built-in tube pump which forms a unit with the so-called canned motor. The pumped medium lubricates the bearings and simultaneously cools the motor.

Table 3.3 Typical pump heads and capacities for various pump types.

Pump Type	Pump head m	Pump Capacity m³/h
Dry running pump		
base mounted pump	> 200	> 1000
In-line pump	> 80	> 600
Wet running pump	< 19	> 76

3.3.2 Energy efficiency

The EU regulation (EC) 640/2009 [73] applies to the electric motors used in **dry-running pumps**. New efficiency classes [113] have been defined, in which the previously best category EFF1 is replaced by efficiency classes IE2 and IE3 in three stages according to Table 3.4. The adjustment of the efficiency classes was carried out in three stages:

Stage 1 From 16 June 2011, all newly sold electric motors must comply with the requirements of class IE2 with the exception of a few designs and areas of application. Pump motors of efficiency class EFF2 may no longer be placed on the market as of this date.

Stage 2 From 1 January 2015, the even stricter IE3 efficiency class applies, initially for motors with a power output between 7.5 kW and 375 kW. Alternatively, such motors must meet the IE2 requirements and be equipped with speed control.

Stage 3 From 1 January 2017, the requirements of stage 2 also apply to motors from 0.75 kW upwards.

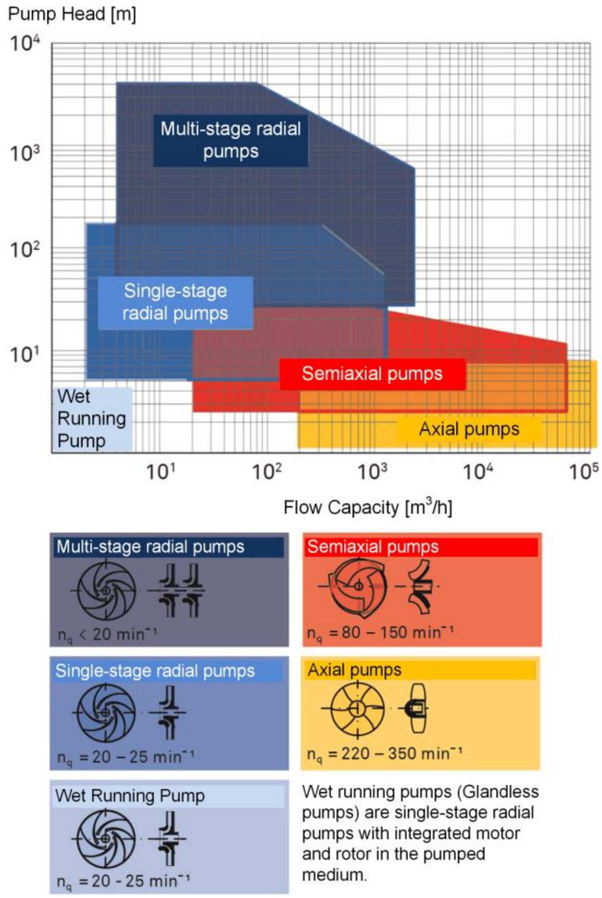


Figure 3.2 Pump typology divided by total head and pump capacity. The rotational speed n_q characterizes the pump design [74].

Table 3.4 Energy efficiency classes for dry running pumps.

Reference Date	Electrical Motors 0.75 kW until < 7.5 kW	Electrical Motors 7.5 kW until 375 kW
June 1. 2011	IE2 or better	IE2 or better
1. January 2015	IE2 or better	IE4 IE3 or IE2 with Speed Control (FU)
1. January 2017	IE4 IE3 or IE2 with Speed Control (FU)	IE4 IE3 or IE2 with Speed Control (FU)

The use of **wet-running pumps** has been largely unregulated although they have a high energy consumption. The basis for the use of future glandless pumps is the Energy Efficiency Index (EEI) according to regulation (EC) 641/2009 [72]. This involves comparing the power consumption within a load profile with a reference pump. The implementation of the new directive is to be carried out in three stages according to Table 3.5. Drinking water circulation pumps are an exception to this rule.

Table 3.5 EEI-Limiting Values for wet-running pumps

Reference Date	External Pumps for Heating and Cooling	Pumps integrated to products	Replacement of integrated Pumps*
January 1. 2013	0.27	no Requirements	no Requirements
August 1. 2015	0.23	0.23	no Requirements
January 1. 2020	0.23	0.23	0.23

*for Pumps that have been utilized prior August 1. 2015

3.3.3 Pump control

A distinction is made between unregulated and speed-controlled circulation pumps. Although unregulated pumps often have a higher efficiency at partial load operation than speed-controlled pumps, they are only suitable if the flow rate varies by less than a factor of 2, for example as a heat circulating pump with a maximum power of 400 W [69].

In district heating networks, on the other hand, pumps with **speed control** are generally used. When designing the pumps, it is important to ensure that they have a high efficiency in practical operation. For speed controlled circulating pumps, the design operating point should be in the last third of the pump characteristic curve in order to achieve a wide control range and, as shown in Figure 3.3, to ensure high efficiency at the same time at the partial load. In addition, the most efficient energy class of motors must always be observed.

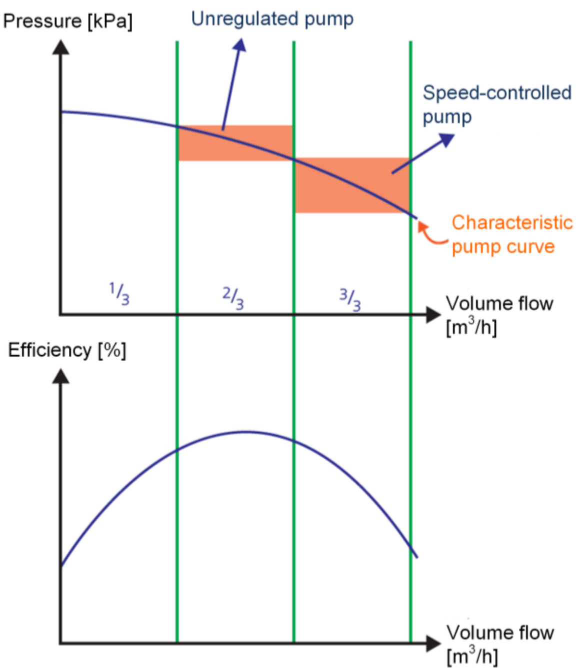


Figure 3.3 Recommended ranges for design of unregulated and speed-controlled pumps [69]. The classification follows a front, middle and rear area (green lines).

Unregulated circulation pumps (Figure 3.4 top)
With uncontrolled circulating pumps the operating point is always on the pump curve. This means that the volume flow and the delivery head do not change. Non-regulated circulating pumps are used when the hydraulic parameters remain constant during a charging process, for example when charging hot water or a main pump with a differential pressure-less distribution.

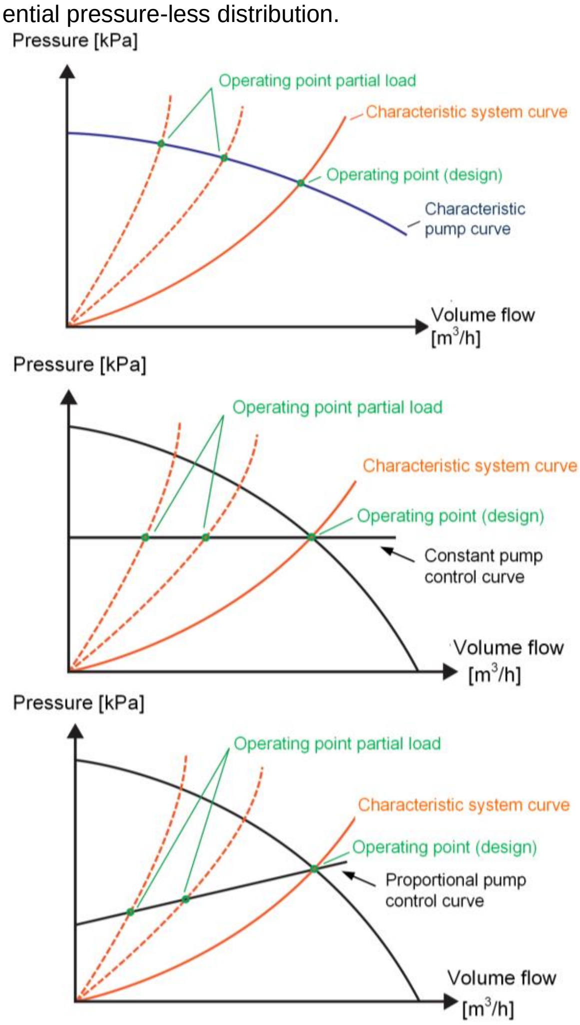


Figure 3.4 Pump diagram showing operation with different control systems [69].
On top: unregulated pump
In middle: constant pressure control
At the bottom: proportional pressure control.

Speed controlled circulation pumps (Figure 3.4 middle and bottom)
Speed controlled circulating pumps continuously adjust the delivery head in the hydraulic circuit in response to changing load conditions. The pump speed is continuously controlled by a frequency converter. The speed of the pump is adapted to the required conditions of the system and adjusts itself for each partial load operating point, thus adjusting the electrical power consumption and significantly increasing the efficiency of the circulating pump. This requires a differential pressure control dependent on the volume flow. The closer the differential pressure measurement is to the relevant consumers, the more efficiently the demand can be met.

For differential pressure control above the pump, a basic distinction is made between the following different types of operation:

- Control with constant operating pressure or also constant pressure control (Figure 3.4 middle) Here the differential pressure over the pump is kept constant with changing volume flow. In the partial load range, the duty point follows the pump control characteristic curve, which is kept constant, horizontally to the left. This operating mode is generally used in smaller heating systems.
- Control with proportional operating pressure or also proportional pressure control (Figure 3.4 bottom) In this control mode the differential pressure across the pump decreases with decreasing volume flow and increases with increasing volume flow. The operating point follows the proportional pump control characteristic in the partial load range, decreasing to the left. As a rule, pumps with proportional pressure control are more efficient than pumps with constant pressure control.

3.4 Pump circuit

In district heating networks, several network pumps are generally used in one pump unit, provided the following conditions are met:

- Parallel connection of two pumps, if only one pump is in operation at a time. The second pump is used as a reserve pump (redundancy).
- Parallel connection of several pumps, if it is cheaper to achieve the required volume flow (efficiency, costs).
- Series connection of several pumps, if it is cheaper to achieve the required delivery head (efficiency, costs).

Operational safety (redundancy) can be ensured by one or more reserve pumps arranged in parallel (see chapter 3.4.3).

3.4.1 Parallel connection

The parallel connection of pumps is advantageous in heating networks with high volume flow and relatively low delivery head (flat network characteristic curve). The pumps arranged in parallel (including a reserve pump) should be of the same type. They can be operated design-dependent with differential pressure control based on a pressure differential measurement in the network as follows:

- Small design: Speed-controlled operation of a single pump (control range A in Figure 3.5)
- Larger design: Control of both pumps in operation with synchronous speed (control range B in Figure 3.5)

Mixed operation (one pump speed-controlled, the second pump at maximum speed) is not permitted with parallel connection.

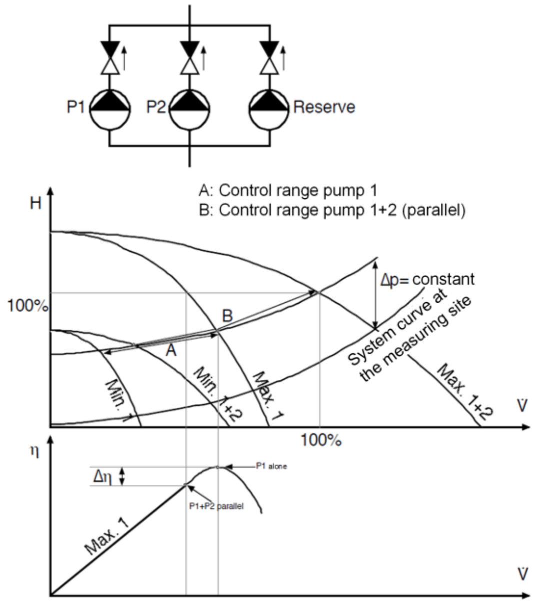


Figure 3.5 Parallel operation of two pumps and one reserve pump [21]. Δp = constant corresponds to the pressure loss across the substation at the relevant heat consumer (usually the most distant heat consumer)

3.4.2 Series connection

The series connection (serial connection) of pumps is advantageous in heating networks with a high delivery head and relatively low volume flow (steep network characteristic curve). For example, a booster pump in the network is also connected in series with the main pump.

If the pumps are connected in series as a unit as shown in Figure 3.6 a bypass with automatic shut-off valve is arranged parallel to each pump. A reserve pump is arranged in such a way that it can take over the function of each individual pump and the pumps connected in series can be switched over mutually. All pumps should have the same pump type.

The differential pressure control with a pressure differential measurement in the network can be carried out design-dependent as follows:

- Small design: One speed-controlled pump alone (control range A in Figure 3.6)
- Larger design: Simultaneous speed-control of both pumps (control range B in Figure 3.6)
- Larger design (as an alternative): Speed control of one pump, the second pump is operated at maximum speed.

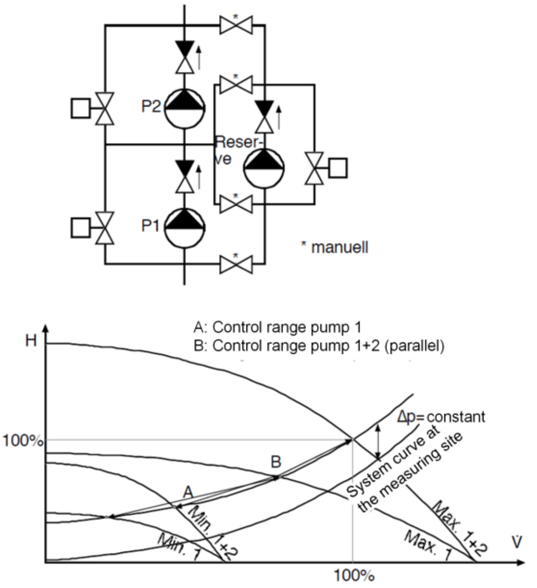


Figure 3.6 Series operation of two pumps and one reserve pump [21]. Δp = constant corresponds to the pressure loss across the substation at the relevant heat consumer (usually the most distant heat consumer).

3.4.3 Redundancy

The pumps are of central importance for the operation of a district heating system. Therefore, a redundancy of the network pumps is recommended. This allows the network to continue to operate if one pump fails. The following redundancy settings are common:

- 2 x 100 % of the nominal capacity
- 3 x 50 % of the nominal capacity
- 4 x 33 % of the nominal capacity.

The redundancy settings of 3 x 50 % or 2 x 100 % etc. are the subject of an economic efficiency analysis. 3 x 50 % or 4 x 33 % are particularly suitable for large networks, as investment costs for large pumps increase disproportionately and operation and maintenance of smaller pumps are easier. In addition, networks under development often require only a fraction of the pumping capacity of the full extension during the first years of operation. This means that a pump group with 2 x 33 % pumping capacity can often be installed for the first few years. This reduces the initial investment and allows the pumps to be operated more efficiently. Additional pumps can be installed later.

For small networks, a cost efficient 2 x 80 % redundancy setting can be used, so that if a pump fails, 80 % of the pumping capacity is available.

3.5 Network control

In a district heating network, the indirectly connected heat consumers require a differential pressure via the heat transfer station. This minimum differential pressure is usually between 0.4 and 1 bar. The network is regulated in such a way that the minimum required differential pressure between flow and return flow is available at each heat transfer station.

At one point in the network there is a minimum differential pressure between the supply and return flow. This point is called the network critical point or, in short, the critical node. This point can move in the network as a function of the current heat demand.

The differential pressure is supplied by the network pumps. Various concepts are available for controlling the pumps:

- Speed control of district heating pumps with differential pressure measurement over the pumps and control using a constant or proportional pump control characteristic.
- A differential pressure measurement can be installed at one or more reference measuring points in the network, which act on the speed control of the district heating pumps in the central control station. Reference measuring points are points in the network remote from the central unit (critical node).
- As an alternative or as a command variable for a master controller, the valve position of the primary valve (e.g. pressure independent control valve) can also be recorded at selected transfer stations (with a control system, for example, the valve positions of all primary valves can be recorded and evaluated). The evaluation of the valve position is then used to control the network pump to the required differential pressure. It is important that the minimum differential pressure (valve) is guaranteed in any operating condition.

In star-shaped distribution networks, a separate pressure regulation per network branch is recommended. An individual pressure regulation is re-aligned for each network branch. This prevents excessive differential pressures and increased power consumption of the pumps. However, a separate pump group must be installed for each branch.

For network arrangements with several feed points, one feed point must be declared as master for the pressure regulation. The second and each further feed point delivers a certain heat output into the network according to specifications. The algorithm for feeding several points into the network must also take into account that no undersupply due to flow standstill occurs between the feed points.

3.5.1 Control concept

As shown in chapter 3.2, network operation is usually based on weather conditions and a timer. The preliminary regulation can be implemented using the following:

- Superordinate I&C-System (superordinate instrumentation and control system)
- PLC (programmable logic controller) of the firing system (e.g. wood combustion)
- Separate individual regulators

If several pumps are used, the following points must be observed when switching the individual pumps on and off in a complex control system:

- Minimisation of the pressure fluctuations in the network at the changeover point by adapting the required speed as quickly as possible. The following points partly contradict this requirement.

- Prevention of pump overloads at the changeover point: If pump 1 is operated at maximum speed for a certain period of time, first reduce this speed to the speed required for use with pump 2 and only then increase pump 2 to the required speed.
- Operation in the rear third of the pump curve must be avoided at all costs, as this can lead to damage to the pump due to electrical overload.
- Prevention of frequent on and off switching by means of suitable switching criteria and corresponding time delays.

This switching on and off effects the operating points and efficiency of the individual pumps. In order to operate pumps more efficiently, the following is valid:

- The optimum efficiency is usually in the middle third of the characteristic curve for the highest speed stage. If the operating point is placed in this range, this means for speed-controlled pumps:
  - As long as the control ranges A "Pump 1 alone" and B "Pump 1 + 2" overlap, the better option is "Pump 1 alone" (Figure 3.5 and Figure 3.6).
  - In the case of series connection, it must be clarified on a case-by-case basis whether it is worth designing the differential pressure control for one pump alone and operating the second pump at maximum speed.
- The optimum efficiency should be the aim for the mode with the highest number of operating hours. If "Pump P1 alone" is designed for optimum efficiency at maximum speed in parallel connection, the joint operation of "P1 + P2 parallel" at maximum speed results in a loss of efficiency.
- In order to evenly distribute the wear and tear of pumps of identical design, the operating times should be evenly distributed by switching the priorities.

3.5.2 Measuring point of the differential pressure regulation

The measuring point of differential pressure regulation must be selected in such a way that the pressure difference fluctuation in the network is small enough to ensure trouble-free operation at any operating point without over or under-supply. Depending on the choice of the measuring point, disturbances can be compensated in different ways:

- When measuring the pressure difference in the district heating network, the pressure difference is kept constant parallel to the network characteristic line at the measuring point (Figure 3.7). This results in a pump control curve that is inclined backwards with respect to the pump. As soon as the flow rate in the network decreases, the delivery head of the pump is reduced to a similar extent as the pressure drop in the network decreases and thus the operating point of the pump follows the system characteristic curve. The behaviour is comparable with a proportional pump control curve as shown in Figure 3.4 below.
- With constant pressure or proportional pressure regulation, the pressure difference must be measured by the pump. This results in a horizontal or a backward sloping pump control curve. Thus, only disturbances which are caused by the steepness of the pump characteristic curve are compensated.
- When measuring the pressure difference in the flow and return of the pump in the central heating plant, faults caused by the hydraulic network upstream of the measuring point (pressure drop via fittings, heat meter etc.) are also compensated.

Conclusion:
The speed control of the pumps via a constant differential pressure in the heating network (critical node) is the more efficient but more complex solution than via the pumps with a constant pressure or proportional pressure regulation.

In case of a change in the consumer structure (more or fewer consumers) or of partial operation, the network critical node or the decisive heat consumer can 'move'. Especially when the consumer structure changes, the pump control curve may have to be adapted to the new situation.

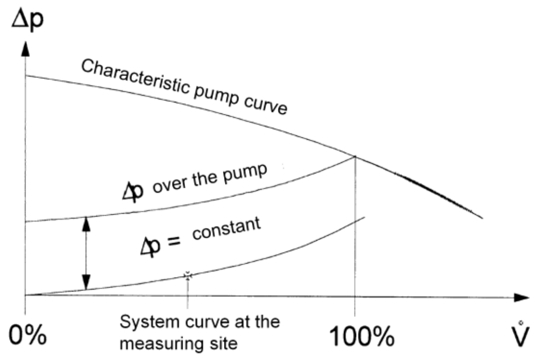


Figure 3.7 Example of a pressure difference measurement in a heating network [21]. Δp = constant relates to the pressure drop at the substation of the relevant heat consumer (e.g. most remote heat consumer).

3.5.3 Heating networks with highly variable flow rate and high supply temperature

The flow rate in large district heating networks is often highly variable, while the supply temperature can only be altered with small changes depending on the outside temperature (see Figure 3.8).

In this case it is advantageous if the two three-way valves can be switched on and off. This results in a flow jump which can be mitigated by a sequence regulator. The two three-way valves connected in parallel are controlled via the sequence regulator with two outputs between 0 % and 100 %. The sequence and the on/off switching of the two three-way valves must be done in the following ways:

- Regulation of the smaller three-way valve via regulator output 1.
- Activation of the larger three-way valve and switching off of the smaller three-way valve when flow rate (signal flow rate measurement heat meter) > flow rate at

maximum specified pressure drop across smaller three-way valve.

- Regulation of the larger three-way valve via regulator output 2.
- Activation of the smaller three-way valve if flow rate (signal flow rate measurement in heat meter) > flow rate at maximum specified pressure drop across the larger three-way valve.
- Regulation of both three-way valves together via regulator output 2.
- Switching off the smaller three-way valve when the flow rate (signal flow rate measurement in the heat meter) < flow rate at minimum specified pressure drop across both three-way valves.
- Regulation of the larger three-way valve via regulator output 2.
- Switching off the larger three-way valve and switching on the smaller three-way valve when the flow rate (signal flow rate measurement in the heat meter) < flow rate at minimum specified pressure drop across the larger three-way valve.
- Regulation of the smaller three-way valve again via regulator output 1.

Alternatively, the three-way valves can be switched on and off when a pressure difference across the three-way valves is detected, using the same logic as described.

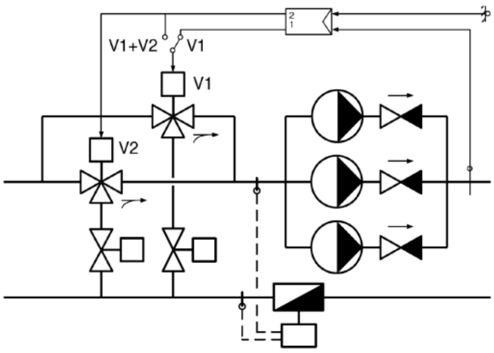


Figure 3.8 Parallel operated network pumps with two control valves to enhance the minimum regulated heat capacity [21].

3.5.4 Separated pump group for winter and summer operation

For an optimal summer operation with a considerably smaller volume flow (less than 20 % of the total volume flow), the separate operation of a parallel speed-controlled summer pump is advantageous (see Figure 3.9):

- The engineering can be simplified (no sequence, but manual switching between summer and winter operation).
- The pump for winter operation should be able to supply 100 % of the required heat at the designed point (no parallel connection of two pumps, possibly install a replacement pump).
- The three-way valve and the pump for summer operation can be dimensioned according to the requirements in summer.
- When pre-setting a distribution line for summer operation, the installation of two valves is recommended if the maximum summer heat output requirement is ≤ 10 % of the maximum winter heat load demand.

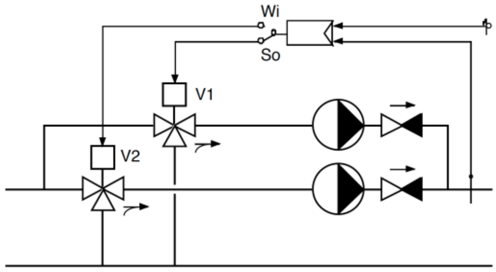


Figure 3.9 Parallel operated summer and winter pumps with two control valves in order to enhance the minimum controlled heat capacity [21].

3.6 Nominal pressure

Pipelines and components such as flanges, valves, gate valves and other fittings are specified in the SN EN 1333 [99] in certain nominal pressure stages, which are designated PN for "Pressure Nominal". The nominal pressure stage determines the minimum thickness and dimensions of pipes, flanges, valves, fittings, etc.

For the design of all pressurised pipes and components, the maximum permissible operating and test pressure must be respected. The PS (Pressure Superior) or MAOP (Maximum allowable Operating Pressure is specified in the Pressure Equipment Directive [66]. The maximum permissible operating pressure is determined by the manufacturer of the component. However, more complex pipe systems are not covered by the Pressure Equipment Ordinance which only describes pressure equipment found in pressure regulating and compressor stations.

The Pressure Equipment Directive [66] corresponds analogously to the European Pressure Equipment Directive 97/23/EC (PED), on the basis of which guidelines [68] were drawn up. With regard to the term standard pressure equipment and with reference to Article 1 (3) (b) of the Pressure Equipment Ordinance [66], Guideline 1/17 makes the following statement: A standard pressure equipment is not specially designed and manufactured for a specific conveyance pipeline, but is intended for use in a number of applications, including other conveyance pipelines or, for example, industrial piping.

Typical examples of standard pressure equipment annexed with pipelines, pressure reduction stations or compression stations may include: measuring devices, valves, pressure regulators, safety valves, filters, heat exchangers, vessels. Such equipment is covered by the directive.

District heating networks with steel medium pipes (pre-insulated rigid steel pipes and pre-insulated flexible steel pipes) are, unless otherwise specified, set at PN 16, although the pipes can be used up to PN 25. Components (fittings, valves, etc.) up to DN 150 are either identical in design to PN 16 or not significantly more complex or cost-intensive than those with a lower pressure rating. With larger nominal diameters or higher nominal pressure ratings (e.g. PN 25) the component prices increase. Domestic installations and heat transfer stations in higher pressure stages than PN 16 are therefore more expensive and also more complex in operation.

Important criteria for the selection of the nominal pressure are the geodetic conditions, the maximum flow pressures and the required pressure overlay of the network.

District heating networks with plastic media pipes have a low-pressure rating of PN 6 and are also only available in relatively small nominal diameters (max. DN 150).

In individual cases a higher pressure than the nominal pressure (PN) can be chosen, if the external boundary conditions are known and a calculation demonstrates, that the maximum allowed operating pressure (MAOP) will be met. Otherwise, the nominal pressure (PN) of the pipes and fittings is the maximum allowed operating pressure (MAOP).

3.7 Pressure curve in the district heating network

The pressure diagram shows the pressure conditions in the network as a function of the distance between the heating plant (from the network pump) and the critical node (the farthest consumer). In the example in Figure 3.11, the pump is installed in the supply pipe. The pressure control is located in the return flow and ensures a constant pressure in the central heating plant (pre-pressure control). As the output changes, the mean pressure in the network and thus the pressure curve fluctuates at all heat transfer stations. The pressure diagram is divided into different sections:

- **Test pressure**: The pressure at which the entire district heating network or sections of it are tested for leaks. The test pressure usually corresponds to 1.3 times the maximum permissible operating pressure and is applied over a period of 24 hours.
- **Maximum Incidental Pressure (MIP):** Maximum pressure allowed by safety devices that can occur in a system for a short period of time.
- **Maximum Allowable Operating Pressure (MAOP):** At no point in the network may the pressure exceed the maximum allowable operating pressure of the pipeline and the installed plant components during normal operation (Maximum Operating Pressure). The maximum allowable operating pressure depends on the nominal pressure rating and depends on network length, operating temperature and geodetic conditions in the network. For cost efficiency reasons this should be as low as possible. Typical nominal pressure ratings for district heating networks are PN16 to PN25. Pipes and in particular system components from PN25 upwards increase the price of the network considerably and should only be used if the conditions make it absolutely necessary (see also Section 3.6 Nominal pressure). Malfunctions in network pressure control can lead to excessive pressures. For pumps which can exceed the maximum allowable operating pressure at maximum delivery head and static pressure, it is recommended to use a safety pressure switch which directly locks the network pump(s).
- **Pressure curve supply flow:** The pressure curve of the supply flow from the heating plant to the house connection, or critical node (corresponds to the pressure drop due to pressure loss in the pipes and fittings).
- **Differential pressure substation:** The pressure difference between supply and return flow at the end of the house connection is defined by the minimum necessary differential pressure across the substation. The minimum differential pressure for a substation can be 0.7 to 1.0 bar.
- **Pressure curve return flow:** The pressure curve of the return flow from the house connection to the heating plant (corresponds to the pressure drop due to pressure loss in the pipes and fittings).
- **Pressure loss in the network:** The dynamic pressure changes depending on the load and is to be compensated by the pump. It is made up of the losses in the pipes and the minimum differential pressure of the relevant heat consumer (usually the furthest away heat substation). The maximum dynamic pressure can be expected at the design point.
- **Static pressure:** The static pressure is the difference between the highest and lowest point (height difference) in the district heating network (differential water column).
- **Minimum operating pressure:** Consists of the static pressure and a safety margin to prevent negative pressure formation, evaporation and cavitation. This means that at the highest point of the network the pressure of the district heating water must be at least 0.5 bar higher than the steam pressure at the maximum network temperature (see Figure 11.1). An additional 1 bar is recommended for any unexpected events. Depending on the situation, this can correspond to the static pressure of the pressure control.
- **Total network pressure:** The maximum network pressure (static and dynamic components) is the pressure at maximum load at a specified point and at maximum network temperature. In this case the maximum network pressure must not exceed the maximum operating pressure.

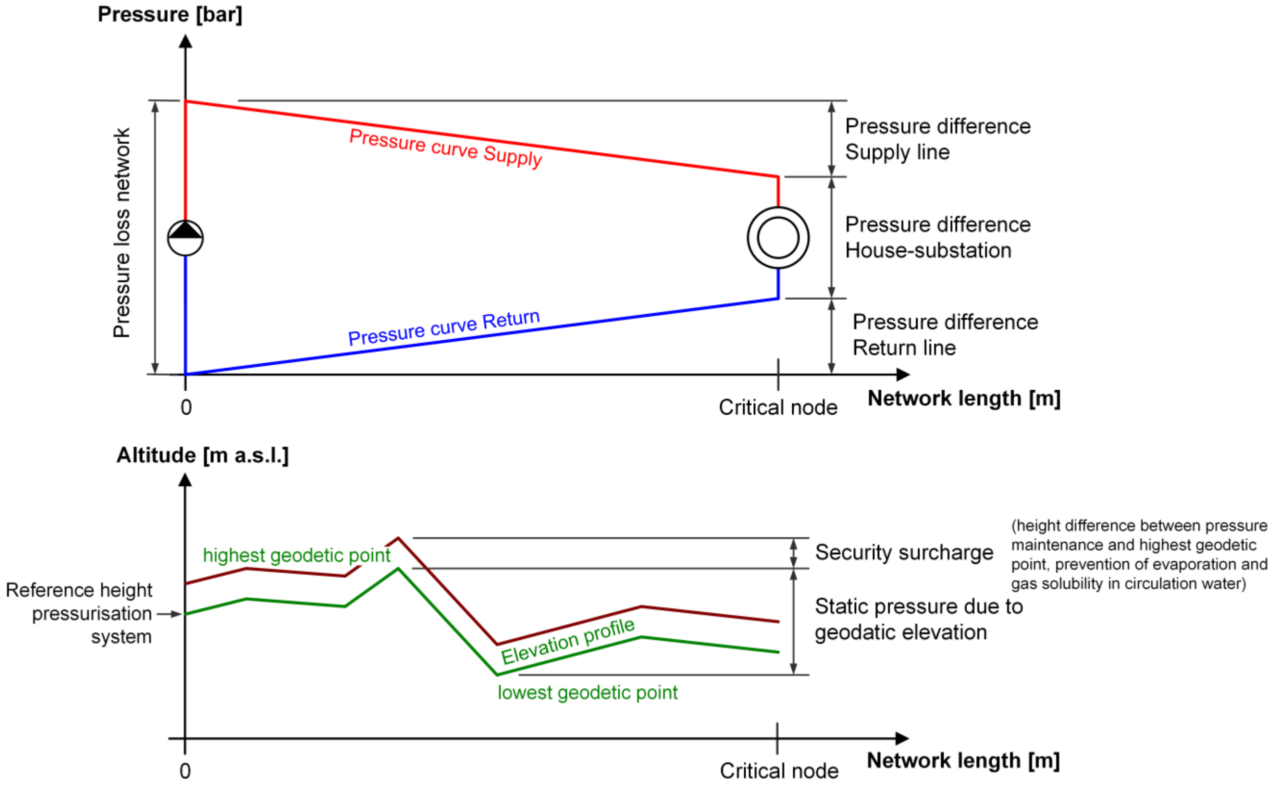


Figure 3.10 Qualitative representation of the network characteristic curve (top) presented as pressure in bar and the static pressure (below) as elevation curve in meter above sea level as a function of network length to the critical node.

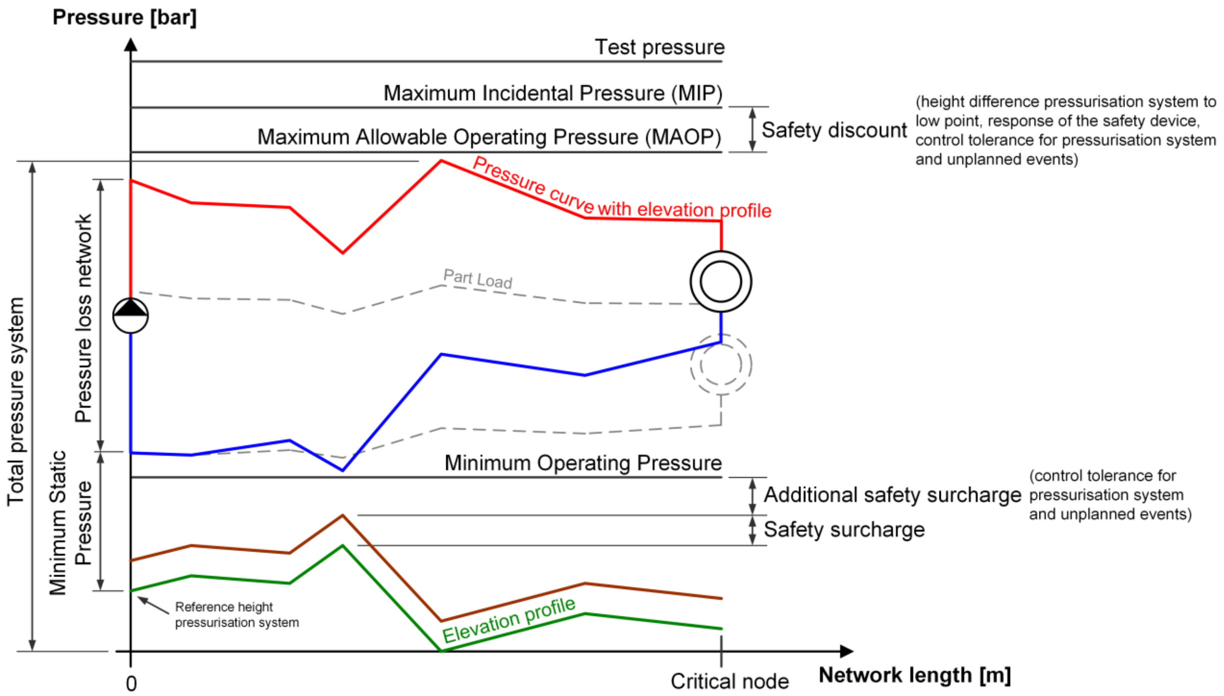


Figure 3.11 Stacked pressure diagram for a heating network with the superposition of the height curve and the characteristic curve of the network as a function of the network length to the critical node.

3.8 Pressure boosting and network seperation

For larger and extended networks, an economic analysis is recommended to determine whether the pressure loss in the network should be provided by the central transmission pumps (nominal pressures PN 25 or even PN 40 instead of PN16), or whether decentralised booster stations or network separations are advantageous. The use of booster stations and network separations can save operating and investment costs, depending on the situation.

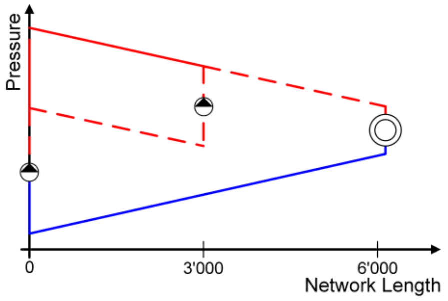


Figure 3.12 Example pressure curve with and without booster pump.

As shown in the example in Figure 3.12, a **booster pump** with a correspondingly lower capacity only delivers part of the volume flow to the furthest heat consumer. This example shows that with a booster pump there is a lower pressure demand than without a booster pump.

A **network separation** describes a hydraulic separation of two otherwise connected systems in district heating supply. Possible reasons for disconnection from the grid are:

- Protection against excessive pressures (house connection, geodetic height difference, pipe length, etc.)
- Separation for different media (e.g.: steam-water or thermal oil-water)
- Other network parameters (pressure and temperature)
- Secondary networks.

The resulting loss of exergy during heat transfer must be considered as disadvantage. This should be kept as low as possible by design and imply a minimum least temperature difference of the heat exchanger.

3.9 Pressure control

With the change in temperature, heat transfer media undergo a change in volume depending on the volume expansion coefficient. Water has the highest density at 4 °C and thus the smallest volume, so that an increase in volume occurs with increasing temperature. In the temperature range between 4 °C and 30 °C, however, this increase is only slight, whereas it is more pronounced at higher temperatures. In a closed system, the volume change must therefore be compensated for in order to keep the pressure constant. The expansion and pressure control systems used for this purpose are therefore of central importance in heating and cooling circuits and fulfil three basic tasks:

- To keep the pressure at any point in the system within permissible limits, i.e. not to exceed the maximum permissible operating overpressure, but also to ensure a minimum pressure to prevent cavitation (negative pressure, evaporation, gas solubility).
- Compensation of volume fluctuations of the heat transfer medium (water) due to temperature fluctuations.
- Provision of system-related water losses in the form of a water reserve.

Careful calculation, commissioning and maintenance are basic requirements for the correct functioning of the pressure control. Further information and recommendations for implementation can be found in the AGFW Worksheet FW 442 Pressure control in heating water district heating networks [104]. The individual pressure control systems, the integration of pressure control as well as the nominal volume and the make-up mass flow of the expansion system are described below.

3.9.1 Pressure control systems

There are different pressure control systems. Each requires an expansion tank and a system for maintaining the static pressure (e.g. compressor for gases or pump for liquids). Pressure control with an open or closed vessel is called static pressure control, with a pressure dictation pump it is called dynamic pressure control.

As a rule, the pressure control systems must ensure that reserve capacities are always available.

Pressure control with open vessel
In the case of pressure control with a high open vessel, the vessel is set up at an elevated location in the network as an equalizing tank. The pressure-maintaining propellants in this system are:

- the air pressure which presses on the water surface of the open vessel
- the weight of the water in the expansion tank.

The advantage of this principle is the low-cost design, as practically no fittings, control devices and pumps are required. Problems with the high open vessel are the occurrence of corrosion in the district heating network due to the permanent entry of oxygen via the venting pipe and cooling of the water. The challenge of finding a sufficiently high location for the installation in order to ensure the appropriate pressure is also often difficult. Today, pressure control with an open vessel is largely avoided.

Pressure control with closed vessel
Gases can be compressed and used as pressure retaining propellants in a closed vessel. This property is used for pressure control in district heating networks. If water losses occur in the network, the compressed gas cushion in the expansion tank presses the water surface. When there is a shortage of water in the district heating network, the pressure on the water surface feeds the required amount of water from the expansion tank into the network. Due to the risk of corrosion by air, an inert gas such as

nitrogen is usually used as a pressure-maintaining propellent. A further measure is to separate the two media by a membrane (membrane expansion vessel MAG).

Pressure control with pressure dictation pump
The pressure dictation pump is installed between the container and the district heating network. In case of water shortage and the resulting pressure drop in the district heating network, the pressure dictation pump pumps the missing amount of water from the container into the network. A backflow of the district heating water is prevented by a non-return valve. When the pressure in the district heating network increases, an overflow valve opens so that the water can flow from the network into the container.

3.9.2 Integration of pressure control

The hydraulic integration of pressure control into the system has a fundamental influence on the working pressure which is composed of the static pressure level of the pressure control and the differential pressure which is generated when the circulating pump is running. A basic distinction is made between three types of pressure control:

- - Pre-pressure control
- - Follow-up pressure control
- - Medium pressure control

In practice, however, other alternatives are also used.

Pre-pressure control
The pressure control is integrated before the circulating pump, i.e. on the suction side. For this reason, the term suction pressure control is also used. This type is used almost exclusively because it is the easiest to control.

Advantages:
- Low static pressure level
- Working pressure > static pressure, therefore no danger of negative pressure formation.

Disadvantage:
- at high circulating pump pressure (large network) high working pressure, observe Maximum allowable operating pressure (MAOP)

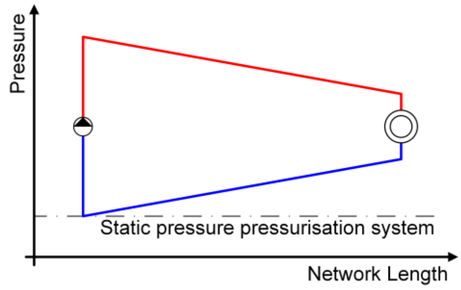


Figure 3.13 Pre-pressure control.

Follow-up pressure control
The pressure control is integrated after the circulating pump, i.e. on the pressure side. When determining the resting pressure, a system-specific differential pressure component of the circulator pump (50 % to 100 %) must be taken into account. The application is limited to a few applications such as solar systems.

Advantage:
- Low static pressure level if the entire pump pressure does not have to be loaded.

Disadvantage:
- high static pressure level
- particular attention must be paid to maintaining the required inlet pressure according to the manufacturer's specifications for the circulating pump, otherwise there is a risk of cavitation

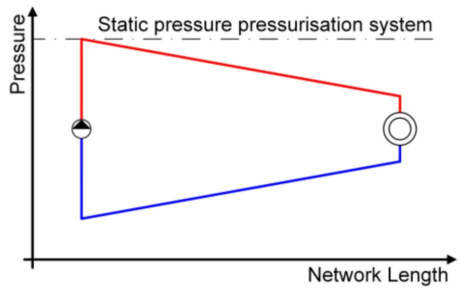


Figure 3.14 Follow-up pressure control.

Medium pressure control
The measuring point for the resting pressure level is "moved" into the system by an analogy measuring section. The resting and working pressure levels can thus be optimally matched and variably designed (symmetrical, asymmetrical medium pressure maintenance). Due to the relatively high technical complexity of the apparatus, its use is limited to systems with complicated pressure conditions, mostly in the district heating sector.

Advantage:
- optimal, variable coordination of working and resting pressure.

Disadvantage:
- high apparatus-technical expenditure

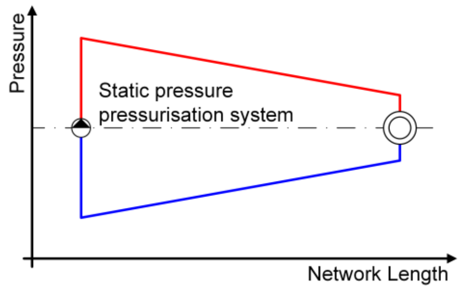


Figure 3.15 Medium pressure control.

3.9.3 Expansion volume and make-up

The relevant parameters of the expansion system and pressure control of a district heating plant are the nominal volume and the make-up capacity of the expansion system.

The nominal volume of the expansion system or the expansion vessel is made up of the theoretical expansion volume and the additional water supply to cover leakages. Because the pressure in the expansion tank is usually generated by a gas cushion and the water level and pressure in the gas space are related (p x V = constant), it is not possible to use the entire volume of the expansion tank for water replenishment. The nominal volume is therefore increased by a factor

$$V_N = \left(\Delta V + V_{MuW}\right)\frac{p_{tot} + 1}{p_{tot} - p_0}$$

The theoretical expansion volume of a district heating system is the make-up volume when energy is removed or the overflow volume when energy is added due to the temperature change of the heat transfer medium. The make-up volume is calculated as follows:

$$\Delta V = V_0\left(\frac{\rho_0}{\rho(T_x)} - 1\right)$$

It is important that the water content V_0, which is necessary for the calculation of the make-up volume, is recorded correctly (see calculation example). This includes the total water volume in the closed system to which the expansion system is connected (heat exchanger, energy storage tank, boiler, piping, intermediate vessel expansion, etc.). The water volume compensates for any water losses in the system and should be approximately 0.5 % of the water content V_0.

$$V_{MuW} = 0.005\ V_0$$

The make-up mass flow of the expansion system depends on the heat load supplied or removed, the volume change coefficient and the specific heat capacity of the heat transfer medium. The make-up mass flow is necessary for the design of the overflow valve and/or the pressure booster pump and is calculated as follows:

$$\dot{M} = \gamma(T_x)\frac{\dot{Q}}{c_p}$$

The required volume change coefficient to calculate the make-up mass flow depends on the temperature and density difference of the heat transfer medium and is calculated as follows:

$$\gamma(T_x) = -\frac{\Delta\rho}{\rho_0}\frac{1}{\Delta T} = -\frac{\rho(T_x) - \rho_0}{\rho_0\left(T_x - T_0\right)}$$

The following points must be taken into account when designing the expansion and pressure-maintaining device:

- required heating capacity
- supply and return temperature, maximum temperature and filling temperature
- pressure conditions in the system
- medium (water, water-glycol, other).

Depending on the application and size of the plant, an expansion and pressure control device with fixed gas filling (small plants), with compressors (precise pressure maintenance possible) or with pumps (transfer pumps for large plants) is recommended.

A pressure fluctuation of ± 0.2 bar is regarded as sufficient control quality in a network. Within this range, the compressors or transfer pumps must be able to feed into the network.

In the following calculation example, the expansion volume and from this the nominal volume as well as the theoretical make-up mass flow of the expansion system (pre-pressure maintenance) is calculated for a district heating network with 1 MW generation capacity.

V_N	= Nominal volume [m^3]
ΔV	= Make-up volume [m^3]
V_{MuW}	= Volume make-up water [m^3]
V_0	= Water content [m^3]; entire water volume within a closed system
p_0	= Minimum static pressure [bar]; to avoid low pressure generation, vaporization and cavitation.
P_{tot}	= Entire pressure network [bar]; for design reasons the pressure at the highest temperature is essential.
$\rho(T_x)$	= Medium density for the end-temperature [kg/m^3]
ρ_0	= Medium density for the initial temperature [kg/m^3]
$\dot{M}$	= Mass flow make-up water at expansion (kg/s)
$\gamma(T_x)$	= Volume change coefficient [1/K]
T_x	= End-temperature [°C]; normally the higher temperature (e.g. maximum supply temperature)
T_0	= Initial-temperature [°C]; normally the lower temperature (e.g. filling temperature)

Pressure rates will be classified as overpressure.

Calculation Example Expansion Volume and Make Up Quantity

Initial Position – Initial Parameters for the Calculation Example

	Temperatures:	Density (based on Table 11.1)
Filling Temperature:	T_F = 10 °C	ρ_F= 1000 kg/m³
Maximum Temperature Supply:	T_{SPLY} = 75 °C	ρ_s = 975 kg/m³
Maximum Temperature Return:	T_R = 45 °C	ρ_R = 990 kg/m³
Safety Limit:	T_S = 90 °C	ρ_S = 965 kg/m³

Specific Heat Capacity (assumed as constant)	c_P = 4.205 kJ/(kg K)
Total pressure for a network at maximum supply temperature	p_e = 12 bar
Minimum operating pressure (pressure maintain)	p_0 = 3 bar

Calculation nominal volume expansion vessel

Water Content V_0

The calculation requires the correct recording of the *Water Content V_0*. This includes the entire water volume within the closed system to which the expansion plant is connected. Find a presented listing as example.

Boiler 1 MW		1 Pc.	3’500 L	=	3’500 L
Pipelines heat generation DN 80		100 m	5.35 L/m	=	535 L
Heat storage		1 Pc.	27 m³	=	27’000 L
Heat exchanger heat generation		0 Pc.	0 L	=	0 L
Intermediate expansion vessel		0 Pc.	0 L	=	0 L
Pipe Lines supply	DN 100	150 m	9.01 L/m*	=	1’352 L
Pipe Lines return	DN 100	150 m	9.01 L/m*	=	1’352 L
Pipe Lines supply	DN 65	350 m	3.88 L/m*	=	1’358 L
Pipe Lines return	DN 65	350 m	3.88 L/m*	=	1’358 L
Pipe Lines supply	DN 25	265 m	0.64 L/m*	=	170 L
Pipe Lines return	DN 25	265 m	0.64 L/m*	=	170 L
Heat exchanger heat consumer		10 Stk.	250 L	=	2’500 L
			Water Volume V_0 Total	**=**	***39'295 L***

*Specific volume of the pipelines in L/m according to Table 13.1

Expansion Volume (Fill temperature- until maximum supply temperature)

The calculation of the expansion volume follows the rule of the maximum supply temperature and results in a volume of 1’008 Litre.

$$\Delta V = V_0\left(\frac{\rho_0}{\rho(T_x)}-1\right) = 39'295\ \text{L}\ \left(\frac{1'000\ \frac{\text{kg}}{\text{m}^3}}{975\ \frac{\text{kg}}{\text{m}^3}}-1\right) \approx 1'008\ \text{L}$$

Make-up water

The calculation of the make-up water results in a volume of roughly 200 Litre.

$$V_{MuW} = 0.005\ V_0 = 0.005 \cdot 39'295\ \text{L} \approx 200\ \text{L}$$

Nominal Volume Expansion Vessel

The calculation of the nominal volume of the expansion vessel results into a volume of nearly 2000 Litre.

$$V_N = \left(\Delta V + V_{MuW}\right)\frac{p_{tot}+1}{p_{tot}-p_0} = (1'008\ \text{L} + 200\ \text{L})\frac{12+1}{12-3} = 1'744.9\ \text{L} \approx 2'000\ \text{L}$$

Calculation mass flow of the make-up water

The calculation example calculates the mass flow of the make-up water by failure of the heat generation yield of 1 MW for a maximum supply temperature of 75 °C to a supply temperature of 65 °C.

Volume change coefficient

The calculation of the volume change coefficient results in a value of 6.12x10^{-4} 1/K.

$$\gamma(T_x) = -\frac{\Delta\rho}{\rho_0}\frac{1}{\Delta T} = -\frac{\rho(T_x)-\rho_0}{\rho_0(T_x-T_0)} = -\frac{975\ \frac{\text{kg}}{\text{m}^3} - 981\ \frac{\text{kg}}{\text{m}^3}}{981\ \frac{\text{kg}}{\text{m}^3}\ (75-65)\ \text{K}} = 0.000612\frac{1}{\text{K}}$$

Mass flow of the make- up water

The calculation of the mass flow rate of the make-up water results in the approximate value of 0.145 kg/s or as a volume flow rate of the approximate value of 0.52 m³/h with a density of 1’000 kg/m³.

$$\dot{M} = \gamma(T_x)\frac{\dot{Q}}{c_p} = 0.000612\ \frac{1}{\text{K}} \cdot \frac{1000\ \text{kW}}{4.205\ \frac{\text{kJ}}{\text{kg K}}} = 0.145\ \frac{\text{kg}}{\text{s}} \approx 0.52\ \frac{\text{m}^3}{\text{h}}$$

4 Heat Distribution – Basics

This chapter describes the individual components, the structure, the installation methods and the typical installation situations when building a district heating network. The design and calculation are dealt with in chapter 7.

4.1 Development

Due to the historical development of district heating distribution technologies, a distinction is made between three established generations [3]. These are described in Figure 4.1 and characterised as follows:

- The first generation is based on the distribution of steam in pipes laid in concrete ducts.
- The second generation is based on the distribution of hot water (> 110 °C) in pipes, which are mainly laid into concrete ducts. Methods with buried and unconnected pipes as well as fillable thermal insulation materials are not popular yet.
- The third and current generation is based on tightly bonded pipes pipes laid directly into the ground to distribute water over a wide temperature range (up to 160 °C in continuous operation).

In addition, for some years now there has also been an interest in distributing heat at low temperature levels that cannot be used directly, which is often referred to as the fourth and fifth generation.

Of particular interest is the development of buried pipe systems, as this is the most widely used technology. In the last century there have been different approaches. Some of them have been used successfully for a long time, while others have not reached the predicted service life or have not met the requirements for energy efficiency.

The main goal of the further development of heat distribution is basically a reduction of:

- Investment costs
- Space requirements
- Installation time
- Operating and maintenance costs.

Some of these parameters are interdependent. For example, reduced space requirements lead to shorter installation times and lower investment costs. An additional effect can be an increased acceptance of district heating if the shorter installation time also reduces the impact on public space.

4.2 Distribution types with decreasing temperature

From steam via hot water to warm water, the historical development of district heating shows a trend towards decreasing supply temperatures and thus decreasing exergy content of the heat (Figure 4.4). This trend is being continued by developments in the distribution of heat at low supply temperatures.

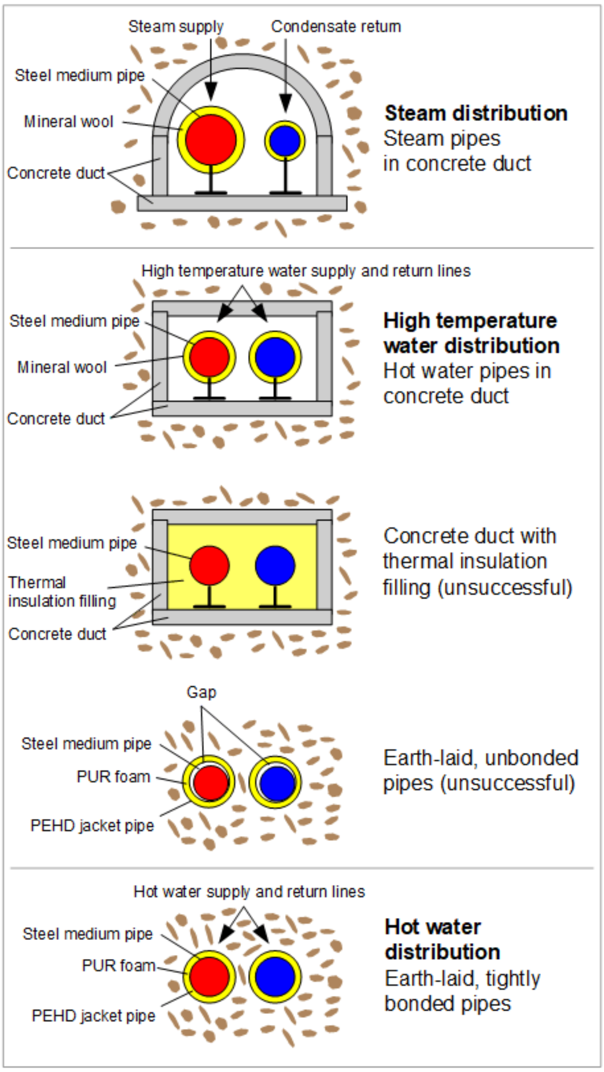


Figure 4.1 Historical development of heat distribution technology. The hot water distribution applies to rigid as well as for flexible pipes in standard and double pipeline configuration (according to [3]).

4.2.1 Steam distribution

Steam was a logical choice as a heat transfer medium over a hundred years ago, as steam was often available from existing steam power plants. The heat utilisation of steam was thus the first application of combined heat and power generation.

Nowadays, water is the dominant heat transfer medium in district heating networks in Europe, although steam networks are still in use. However, these are gradually being replaced by water networks. There is still a niche market for steam networks for applications where a high energy content is required. This applies, for example, to certain industrial processes or, for example, for sterilisation in hospitals.

In Germany, the share of steam in the distribution of heat in district heating networks is only 11 %, with a declining trend [3]. In contrast, steam is often used as a heat transfer medium in the USA, for example in New York in one of the largest district heating systems in the world. There are no plans to replace it with a water network, but no relevant expansions are likely to take place. Steam networks exist in numerous other cities in the USA where occasionally even latest-generation steam boilers are used.

Steam networks are designed as open or closed systems. In an open system, the condensate is drained and must be replaced by fresh water. In a closed system the condensate is returned to the steam generator. The open system has the advantage that no return line is required for the condensate. In contrast, the energy loss in the form of heat is very high at about 15% and replacing the condensate is elaborate.

Although there are advantages of the high energy content of steam and the higher supply temperature than water networks, steam networks have considerable disadvantages:

- The condensing steam in an open system absorbs oxygen and carbon dioxide, which makes the condensed water very corrosive and burdening the return pipe.
- In a closed system, each condenser requires a condensate return device.
- The pressure loss on the steam side is usually much higher than on the water side.
- Cost-intensive devices are required for condensate separation.

4.2.2 High temperature water distribution

The design of the second generation of district heating networks is similar to the first generation except for the difference that water instead of steam is used as the heat transfer medium. The pipes are also laid in ducts. At the beginning of the district heating supply with steam, the ducts were still bricked in and often accessible. Later, the ducts were made much more compact in concrete as shown in Figure 4.1. Walk-in energy ducts (e.g. tunnels) are still used today, for example in large cities, where district heating, district cooling, water and other pipeline-bound energy sources are used in addition to district heating.

Prefabricated concrete ducts accelerate the laying process, whereby valve and compensation shafts often have to be concreted on site. The thermal insulation is applied directly on site in the form of mineral wool. The prerequisite for a functioning thermal insulation is that the mineral wool remains dry, which cannot always be guaranteed in the event of condensation and leaks, as the thermal insulation is not protected against moisture.

To ensure that the rigid pipes are not damaged by thermal expansion and contraction, the pipes are fixed in so-called fixed points. The pipes must be supported and guided between these fixed points. Another possibility for thermal compensation is offered by metal expansion joints or installation with a U-pipe as shown in Figure 4.2.

Experience shows that carefully planned and operated second-generation district heating networks are still in operation and function perfectly.

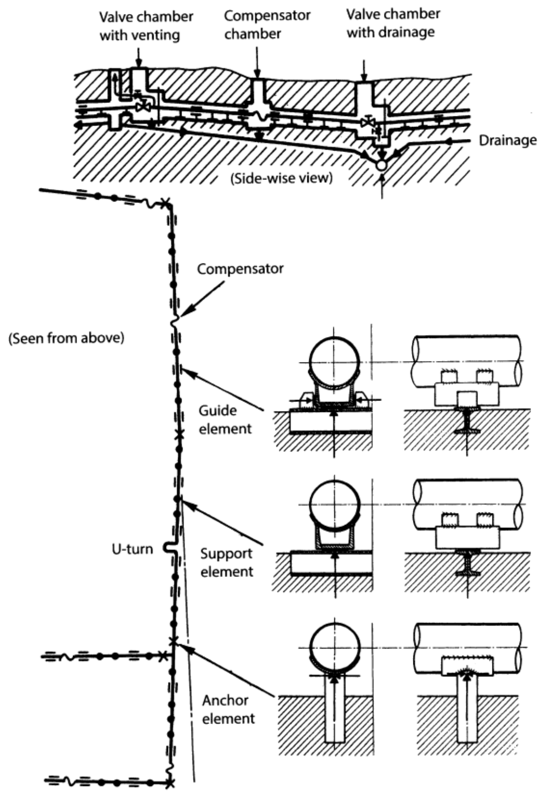


Figure 4.2 Second generation of heat distribution: High temperature water distribution within a concrete duct [3].

4.2.3 Hot Water Distribution

The expensive and time-consuming construction of pipelines in concrete ducts was the main reason for developing underground laying.

Between the second and third generation there were two intermediate steps which finally led to the third generation (Figure 4.1, third and fourth graph from above). The first step was a modification in the installation of ducts, where different filler materials were tested as thermal insulation. The second step concerned the development of plastic jacket pipes laid directly in the ground with a gap between the heat transfer pipe and the thermal insulation to compensate for thermal expansion. These two development steps were unsuccessful due to high costs and deficiencies such as corrosion of the steel heat transfer pipes due to moisture.

Based on these experiences, the plastic jacket pipes were developed further. With the improved method the heat transfer pipe is firmly connected to the heat insulation and the latter is firmly connected to the jacket pipe (Figure 4.3). This largely solved the corrosion problem. Plastic casing pipes are prefabricated and can therefore be laid directly. The design is rigid (as rods) or flexible (as rolls and rods).

Figure 4.3 Third generation of heat distribution: Pre-insulated rigid steel pipe
Source: Brugg Pipe Systems AG

The materials used for the heat transfer tubes are steel, non-ferrous metals and plastics, depending on the application. Today, the thermal insulation consists of a rigid polyurethane foam (PUR), which is produced using a long-term resistant and environmentally friendly propellant gas and has very good insulating properties. The casing pipe must be resistant to practically all chemical compounds found in the ground and at the same time be impact-resistant and break-proof. Seamless extruded pipes made of high-density polyethylene (HDPE) are used for this purpose.

Both rigid and flexible pipes are used in single and multiple designs (e.g. double pipe). In the case of multiple designs, two or more heat transfer pipes are routed in one jacket pipe. However, flexible pipes and multiple pipes are only available in a small diameter range (see section 4.3.1.6).

A special feature of the third generation of heat distribution is that, in addition to the simplified method of laying directly in the ground, fewer components are used to compensate for thermal expansion. While metallic expansion joints are practically no longer used, the elasticity of the pipes is used for thermal compensation and supported by additional bends (changes of direction) in the route.

A further feature of the third generation is the possibility to identify a leakage and its location.

4.2.4 Low Temperature Water Distribution

For the distribution of heat at low temperature level, terms such as "LowEx district heating", "anergy networks" or "cold district heating" are used and it can be part of a more extensive cross-linking, which is also described as "thermal network". Low temperature heat distribution can be used instead of or in addition to a classic district heating network. Although some applications already exist, this further application, also known as fourth generation (shown in Figure 4.4) is still in the development stage. The following approaches, among others, are being pursued:

- An extended application of district heating at a low temperature level
- Heat distribution at low temperature for feeding decentralised heat pumps
- Cross-linking of energy production, energy distribution and energy consumption
- Two-way district heating or prosumer-concept in which a user both uses and supplies energy from or to the district heating network. also act as energy producers.

4.3 Components

4.3.1 Pipe systems

The selection of the pipe system and the suitable installation technique depends on the network temperature and the network pressure as well as on requirements which are largely determined by site conditions. These are:

- Service Lines
- Surrounding area
- Buildings and/or Constructions
- Streets
- Railway tracks
- Underpasses
- Groundwater
- Soil Condition
- Tree Population

Multi-layered metal / plastic composite pipes (pre-insulated rigid steel pipes) are the most commonly used pipe system due to their standardisation, robustness and low price. In addition, flexible pipe systems such as pre-insulated PEX pipes and pre-insulated flexible steel pipes are mainly used in the field of sub-distribution and house connection pipes. Other pipe systems are steel jacket pipes and fiberglass-reinforced plastic pipes (FRP or GRP).

4.3.1.1 Pre-insulated rigid steel pipes

Pre-insulated rigid steels pipes are the most frequently used district heating pipe system. Due to their high resistance to pressure and temperature, they can be used in many situations. They consist of a steel medium pipe connected with an insulation layer of rigid polyurethane foam. The outer casing is made of high-density polyethylene (HDPE) (Figure 4.3). The individual materials are tightly bonded together and is therefore a non-self-compensating pipe system, which is why the thermal expansion is transferred to the entire pipe system.

The maximum continuous load for a service life of 30 years is reliably guaranteed up to a temperature of 120 °C in accordance with SN EN 253 [96]. According to the manufacturer's specifications, continuous operating temperatures of up to 140 °C at a pressure of up to 25 bar are guaranteed. In this case, manufacturers must test the thermal life of 30 years according to Annex C of SN EN 253 [96].

In order to absorb the resulting stresses and strains, complex pipe network statics and suitable compensation measures are necessary. Depending on the requirement profile, pre-insulated rigid steel pipes are installed with or without preheating (cold-laying method 1 or 2) or thermal prestress. The aim of the installation methods is to absorb the thermal expansion of the steel pipes so that the permissible stresses in the steel pipe are not exceeded. The laying method must follow the design and routing as well as the calculation of the pipe network statics. AGFW Code of Practice FW 401 [100] offers simplified planning and helps reduce the detailed mathematical verification to a few special cases. The cold-laying of pipelines up to DN400 is statically approved.

In complicated routes and difficult conditions (factory lines, etc.), many fittings are sometimes required, which makes laying relatively expensive. However, as material costs are low and a high level of operational safety is achieved, pre-insulated rigid steels pipes have become widely accepted as the standard solution, especially for larger diameters from DN 100 upwards.

Pipeline construction is usually carried out by welding the individual pipes together. Certified welders are required as well as a considerable amount of time and effort to inspect the weld seams (X-ray and pressure testing). As an alternative to welding, the pipes can also be joined together with press fitting connections (e.g. Haelok system) for small nominal diameters (up to DN 100).

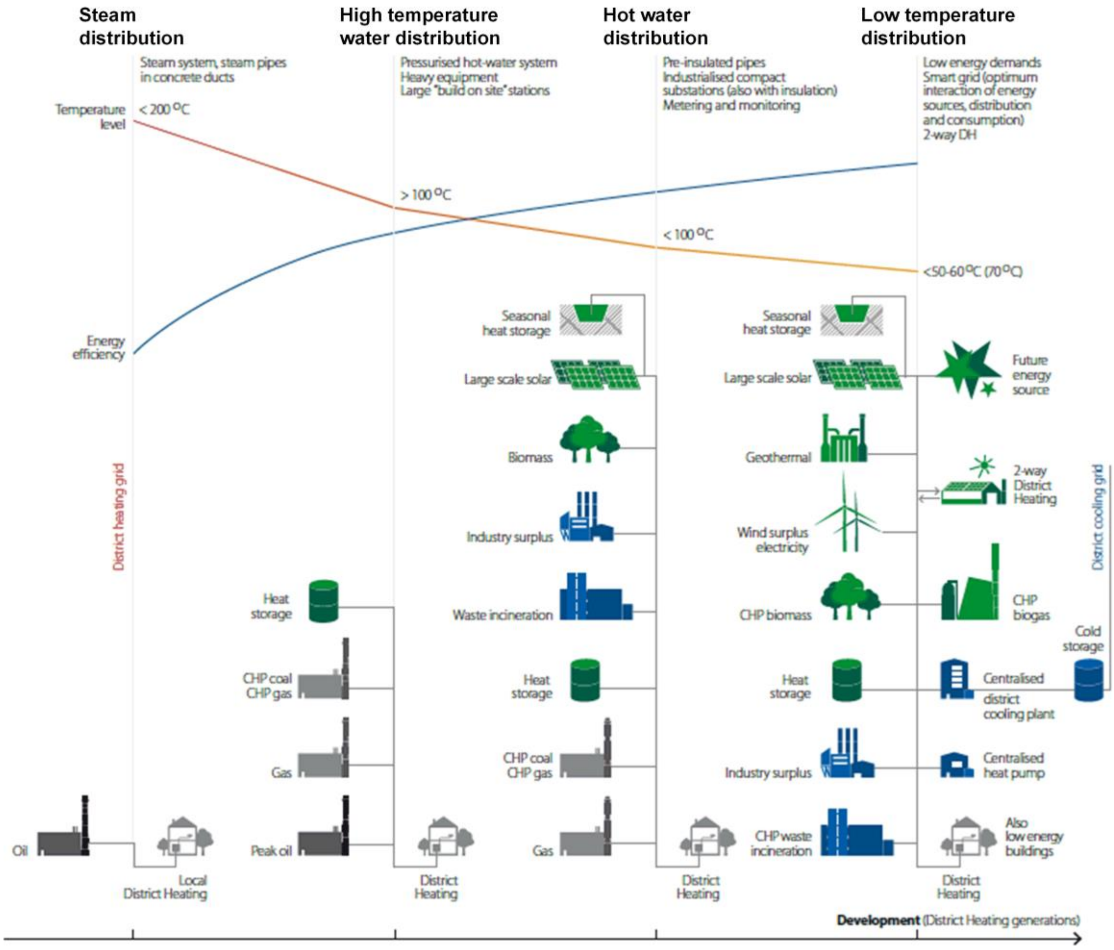


Figure 4.4 Path of development of district heating for four generations with increasing supply of energy sources and increasing complexity with decreasing supply temperature level and increasing energy efficiency (adapted from [42]).

4.3.1.2 Pre-insulated plastic pipes

Pre-insulated plastic pipes consist of a medium pipe made of cross-linked polyethylene (PEX) or polybutene (PB). The thermal insulation and the jacket are similar to those of pre-insulated rigid steel pipes. The medium pipes made of PEX and PB usually have a diffusion barrier for oxygen.

The operating temperature is limited to a maximum of 95 °C and the pressure limit is 6 bar (heating) and 10 bar (sanitary applications, drinking water). For a service life of 30 years, the continuous operating temperature is 80 °C and the maximum pressure is 6 bar. With a limitation to 70 °C and 6 bar, the theoretic service life is extended to 50 years, whereas continuous operation at the upper temperature limit of 95 °C considerably shortens the life expectancy.

Individual pipe sections are connected by means of press couplings, which are assembled with special tools. For pipes made of polybutene (PB), it is also possible to weld the pipes. House connections are realised with T-pieces. However, later connections can only be made by freezing or squeezing the pipe, whereas drilling as with pre-insulated rigid steel pipes is not possible. For this reason, it is recommended to install T-pieces at the beginning for possible later connections

The use of pre-insulated plastic pipes is limited to small to medium district heating networks due to the limited operating parameters of temperature and pressure. An advantage over pre-insulated rigid steels pipes is the simple routing, which can be adapted to local conditions such as topography, pipe crossing or external pipes. In comparison to the pre-insulated flexible steel pipes described below, pre-insulated plastic pipes are cheaper and easier to bend, and they also allow tighter radius. In addition, the requirements for construction and installation personnel are lower.

Because of these advantages, pre-insulated plastic pipes are usually preferred to pre-insulated flexible steel pipes, provided that the lower pressure and temperature resistance allow this. The disadvantages, however, are the relatively expensive fittings, the complex technology of subsequent connection, the fact that no leakage warning systems exists and the higher material costs, which increase disproportionately with the diameter.

4.3.1.3 Pre-insulated flexible steel pipes

In order to ensure the flexibility of pre-insulated flexible steel pipes, the pipes are usually made of corrugated copper or steel pipe or, in the case of small pipe diameters, of soft-annealed straight steel or copper pipes. The thermal insulation and the jacket are designed as for pre-insulated rigid steel pipes.

Pre-insulated flexible steel pipes can be used in continuous operation up to a temperature of 160 °C and a pressure of 25 bar.

The corrugated pipe systems are completely self-compensating and are sold in rolls up to DN 150. The straight pipe systems are self-compensating to a limited extent and are only produced as rolls for small diameters, as the pipe is deformed too much at larger diameters. Both systems therefore require no compensation measures and allow flexible routing. Pre-insulated flexible steel pipes are therefore preferably used for house connections, where flexibility is particularly advantageous and no fittings or junctions are required.

The requirements are the same as for the laying of pre-insulated plastic pipes. However, due to the increased effort required for bending and the higher weight, laying is more complex.

The corrugated profile causes an increased pressure loss, which is why corrugated pipes sometimes have to be designed with a nominal diameter larger than smooth pipes. A further disadvantage is the subsequent connection, which is only possible by using additional T-pieces and these can only be installed when the network is out of operation. However, the simple installation without the necessity of specialised fitters are advantageous. As pre-insulated flexible steel pipes create higher costs than flexible pre-insulated plastic pipes, they are only used when higher pressure and temperature resistances are required.

4.3.1.4 Fibreglass reinforced plastic pipe

A fibreglass reinforced plastic pipe (GRP) consists of a rigid glass fibre reinforced epoxy resin pipe. As with pre-insulated rigid steel pipes, they are insulated by a layer of polyurethane foam insulation combined with a polyethylene jacket pipe. GRP pipes are lightweight and their main advantage is their resistance to corrosion, which is why they are mainly used for corrosive media such as geothermal spring water. The design limit is 160 °C and 16 bar.

The individual pipes and fittings are connected by gluing. Subsequent connections can only be made when the pipe has been emptied, using T-pieces or reinforcing the main pipe with a sleeve and drilling a hole in it. Compared to pre-insulated rigid steel pipes, the pipe network is easier to assess. Depending on the pipe length, concrete collars have to be constructed for changes of direction. In addition, the civil engineering and the dimensions of the trench are comparable to those of pre-insulated rigid steel pipes. Due to the high material price, GRP pipes are only used for applications with special requirements and especially for corrosive media.

4.3.1.5 Steel Jacket Pipe

In the case of steel jacket pipes, unlike pre-insulated rigid steel pipes, the jacket pipe is also made of steel. This allows temperatures of up to over 300 °C and pressures of up to 64 bar. Since steel jacket pipes are expensive, they are only used in very large heating networks with high temperatures or for industrial purposes. The thermal insulation is mainly achieved by a vacuum created in the space between the casing pipe and the service pipes. A layer of mineral fibre is also used to reduce radiation. The vacuum insulation enables significantly lower heat losses than with other systems. The individual pipe sections are welded together and the vacuum in the intermediate space can be monitored to detect leaks in the inner or jacket pipe.

4.3.1.6 Double pipe design

Double pipe designs (also called duo-pipe), along with pre-insulated rigid and flexible steel pipes and pre-insulated plastic pipes, are available in the lower nominal diameter range (Table 4.1 and

Table 4.2). For special applications, the steel jacket pipe can be designed in double or multiple pipe versions. Double-pipe and multi-pipe systems have the following advantages over single-tube systems:

- low installation costs (smaller trench width)
- lower heat loss
- 50% fewer socket joints
- 50% fewer core holes and wall seals (not sure)
- lower number of expansion limbs

In contrast, thermal insulation in a double pipe must ensure that the heat transfer from the flow to the return as well as the return temperature are kept low. Double pipes with pre-insulated plastic pipes or flexible steel pipes are particularly suitable for laying from house to house, as no underground junctions are necessary.

When using double pipes with pre-insulated rigid steel duo pipes, any junctions must be planned carefully as subsequent installation is very costly. The pipe routing must be excavated precisely. Horizontal pipe routing is ideal, as this reduces any movement by one degree. Differences in inclination also require precisely manufactured fittings. Pre-insulated rigid steel duo pipes are interesting for straight pipelines without junctions and with a constant inclination of the pipeline route. In the case of trenchless laying (especially over long distances), a double pipe with a small diameter can be laid.

4.3.1.7 Selection of the pipe system

When realising a district heating network with a central heating plant and distribution network, the investment costs for the distribution network often account for more than 50 % of the total costs. Cost optimisation is therefore crucial for its economic viability, therefore investment and operating costs must be carefully considered. As a first step, the areas to be covered must be defined in order to establish the best possible locations for the heating centre. Based on an appropriate pre-selection for the coverage of the network, the selection and design of the piping system then plays a central role. The main criteria for selecting the pipe system or the pipe systems to be combined in a network are the following parameters:

a) Operating temperature
b) Operating pressure (static and dynamic pressures)
c) Leakage monitoring
d) Heat loss costs
e) Customer density
f) Availability of nominal diameters, in particular maximum pipe diameters
g) Type of civil engineering works
h) Available space for civil engineering
i) Expansion projects

To a: The maximum **operating temperature** of a district heating network determines whether low-cost pre-insulated plastic pipes is suitable or whether steel medium pipes must be used. If operated up to the continuous operating temperature specified by the manufacturers, a service life of at least 30 years is guaranteed.

To b: The maximum **operating pressure** in a district heating network depends on the pressure losses in the network (which must be applied by the network pumps) and on the topography (difference between the highest and lowest point in the district heating network). The choice of material and the pressure level to be selected can be derived from this. In principle, different pressure stages can also be used in a network. In order to clarify whether cost savings can be achieved, the possible operating cases must be compared.

To c: The possibility of **leakage monitoring** must be taken into account when selecting the piping system.

To d: The choice of the **insulation thickness** of the thermal insulation must be clarified by means of an economic efficiency calculation, whereby the capital costs for different insulation thicknesses must be compared to the heat loss costs. The evaluation thus depends, among other things, on the imputed useful life and the interest on capital as well as the heat production costs.

To e: The **density of consumers** and thus the number of branches in the district heating network can have an influence on the choice of pipe system, since, for example, double pipe systems are mainly disadvantageous with a large number of branches due to the complex connection systems.

To f: Attention must be paid to the fact that individual pipe systems only cover a limited **range of pipe diameters** and that in particular the maximum pipe diameter may be limited.

To g: When **laying**, a distinction must be made between different methods. The installation methods are to be examined for the possible use of the selected pipe system and, if necessary, changes are to be made.

To h: If it is known in advance that **space conditions in civil engineering** (especially in underground systems) are subject to restrictions, this must be checked with regard to the selected pipe system.

To i: When selecting the piping system, the final configuration of a project must be considered, as the above-mentioned parameters can change in the course of the **network expansion**. For example, a district heating network could later be operated at higher temperatures and pressures due to an increasing number of consumers or unexpected compression.

Table 4.1on the next page presents an overview of the individual pipe systems.

Table 4.2 lists the standard nominal diameters available for the pipe systems pre-insulated rigid steel pipes, pre-insulated flexible steel pipes and pre-insulated plastic pipes.

Table 4.1 Overview of the different pipe systems.

Pipe system	Range of application				Available lengths			
	Max. permitted operational temperature	Continuous operating Temperature	Nominal pressure PN	Nominal diameter DN	Rod	Rolls	Double pipe up to DN	Characteristic
	°C	°C	bar	–	m	m	–	–
Pre-insulated rigid steel	160	until 140	25	20–1000	6/12/16*	–	DN150	The most commonly used pipe system due to standardization and robustness
Pre-insulated flexible steel	180	until 160	25	20–150	12*	until 1000	DN50	Relatively expensive → justified if the installation conditions make it necessary
Pre-insulated PEX pipes	95	80	6	20–150	12*	until 780	DN50	Relatively inexpensive → limited pressure- and temperature resistance
Fiberglass-reinforced plastic	160	160	16	25–1000	6*	–	–	Relatively expensive → only for special corrosion resistance requirements
Steel jacket	400	400	until 64	25–1200	16*	–	**	Relatively expensive → only if pressure-, temperature or installation conditions makes it necessary

*Standard length, additional lengths are available on request. ** Special designs available on request (e.g. multiple-unit tube)

Table 4.2 Nominal diameters (gray) for the pipe systems pre-insulated rigid or flexible steel pipes and pre-insulated PEX pipes, with information on thermal insulation class and their availability in duo pipe designs.
The compilation includes range of products from the following companies:
Brugg Pipe Systems, Isoplus and Logstor.
TIC: Thermal insulation classes for pre-insulated rigid steel pipes sorted ascending from class 1 to 3.
S: Standard insulation class for pre-insulated flexible steel pipes and pre-insulated PEX pipes;
R: Reinforced insulation class for pre-insulated flexible steel pipes and pre-insulated PEX pipes.

DN	Pre-insulated rigid steel pipe			Pre-insulated rigid steel pipe–Duo			Pre-insulated flexible steel pipe		Pre-insulated flexible steel pipe–Duo		Pre-insulated PEX pipe		Pre-insulated PEX pipe–Duo	
	TIC1	TIC2	TIC3	TIC1	TIC2	TIC3	S	R	S	R	S	R	S	R
20	■	■	■	■	■		■		■		■	■	■	■
25	■	■	■	■	■		■	■	■		■	■	■	■
32	■	■	■	■	■		■	■	■		■	■	■	■
40	■	■	■	■	■		■	■	■		■	■	■	■
50	■	■	■	■	■		■	■	■	■	■	■	■	
65	■	■	■	■	■		■	■			■	■		
80	■	■	■	■	■		■	■			■	■		
100	■	■	■	■	■		■				■	■		
125	■	■	■	■	■		■				■			
150	■	■	■	■	■		■				■			
200	■	■	■	■	■									
250	■	■	■											
300	■	■	■											
350	■	■	■											
400	■	■	■											
450	■	■	■											
500	■	■	■											
600	■	■	■											
700	■	■	■											
800	■	■	■											
900	■	■												
1000	■													

4.3.2 Leakage monitoring

District heating networks can be designed with or without leakage monitoring equipment, depending on the laying technique and pipe system. Electrical monitoring is widely used for pre-insulated rigid and flexible steel pipes and is considered state of the art [5]. Experience has shown that continuous monitoring with centralised leak detection guarantees the network supply and thus minimises damage. For this reason, a **leakage monitoring system is always recommended**. The district heating network should be monitored continuously at designated measuring points.

Leaks can also be located by means of thermography. This method is used if no electrical monitoring was installed when the district heating pipeline was built.

In the case of duct laying, the visual route inspection is considered sufficient and, where appropriate, automated duct monitoring equipment should be used. Above ground level pipelines are usually operated without leakage monitoring systems.

The leakage monitoring systems used today measure either the electrical resistance of the thermal insulation between two cores or between a core and the service pipe. If the thermal insulation or the indicator is wet, the resistance decreases. A monitoring loop can be formed by using two cores. Monitoring this loop ensures that the whole circuit is monitored. When selecting the leakage monitoring system, care should be taken to ensure that the measuring principle allows early detection of damage (moisture penetration of insulation due to damage to the outer jacket or the carrier pipe) and accurate location identification so that maintenance work can begin as quickly as possible.

4.3.2.1 Monitoring systems

The monitoring of a network for leakage can be carried out as follows, for which digital or analog measuring instruments are used:

- **Central monitoring**: All data and values of the monitoring and several monitoring circuits are collected at a central location.
- **Decentralised monitoring**: Monitoring devices are installed on site in the monitoring circuits. Data is either scanned in a periodic cycle or alarm signals are transmitted.
- **Manual monitoring**: The current state values are checked at fixed intervals by portable monitoring devices.

In the case of the monitoring systems described below, the term “system” refers to the wires (cables, cores) embedded in the insulation of the district heating pipe.

Brandes System (NiCr wires)
The so-called Brandes system consists of two wires that do not touch each other: a sensor wire with perforated insulation made of NiCr (80 % Ni and 20 % Cr) and a fully insulated return wire made of copper. The perforation allows the detection of moisture in the foam. The two wires are connected at each end of the pipe to form a monitoring loop. The high resistance of the sensor wire allows the connected pipe length or sensor wire length and thus the location of the moisture to be identified. A further advantage is the uniform application and installation by all manufacturers and suppliers of district heating pipes.

Nordic system (copper wires)
The Nordic system uses two copper wires that do not touch each other. These wires vary depending on the supplier, e.g. bare, tinned and insulated. In recent years, the use of two bare wires has become generally accepted (tinned wires are also considered bare). A major advantage of the Nordic system is the low cost of the wires. However, mixing different types of wires is not recommended. The disadvantages using the resistance reference measuring method described below can be practically eliminated by soldering the connections in the pipe area. As with the fire system, the wires are connected together at all pipe ends to form a monitoring loop.

Hierarchical systems
The hierarchical or indicator system uses two insulated and twisted copper wires in the thermal insulation. Indicators are installed as sensors at the welding points, which for example consist of two copper plates held in plastic nets and insulated from each other by an inorganic ceramic fleece. When moisture occurs, the insulation resistance of the ceramic fleece decreases. The change in resistance is therefore the monitoring criterion. If the entire pipe length is to be checked, corresponding monitoring wires must be installed in the thermal insulation. These wires are then no longer isolated. This system consists of a maximum of four hierarchies to which joints are connected via so-called separators. The number of joints is unlimited. Network expansions can be carried out without interfering with an existing system. All connected pipes are monitored for moisture and the main pipeline is additionally monitored for wire breakage. This system can also be used for service pipes that are not made of metal. Disadvantages are the complex documentation of the different hierarchies, the use of electronic components (susceptible to overvoltage) and the fact that the pipes are only monitored for moisture at the joints.

4.3.2.2 Locating procedure

Resistance reference measuring method
With the resistance reference measuring method (also known as resistance measuring method or resistance comparison locating), locating moisture is carried out according to the principle of the unloaded voltage divider and is mainly used in the Brandes system. Under certain circumstances, however, it can also be used in the Nordic system. A "measuring bridge" is used to compare the two sections - loop start to fault location and loop end to fault location - and thus determine the location of the fault. The accuracy of the location increases with the intensity of the moisture and the decreasing length of the measuring section. The disadvantage is that multiple moisture damages in a test section are not recorded as such. In this case only an average value is given.

Impulse-Runtime Measuring Method
The impulse runtime measuring method uses the wave resistance of the monitoring wires, as is also used for fault detection in cable technology. Pulses emitted by the locating device are reflected to varying degrees at points with deviating "resistance", i.e. moisture or wire interruptions. The fault location is determined by the running time required by the impulse from the application to the fault location. This method requires a higher moisture level than the resistance reference measuring method to locate a leak. The impulse transit time measurement method is well suited for locating wire interruptions.

The most important features of the individual leakage monitoring systems are shown in Table 4.3.

Table 4.3 Comparison of the systems used for leakage monitoring

Criteria	Nordic-System	Indicator-System	Brandes-System (NiCr)
Monitoring wires	Two non-touching and non-insulated copper wires	Two insulated and twisted copper wires with indicators as sensors at the welding points	One NiCr wire with perforated teflon insulation One copper wire
Measurement method	Impulse-runtime-measurement method and Principle of the unloaded voltage divider	Impulse-runtime-measurement method	Principle of the unloaded voltage divider and Impulse-runtime-measurement method
Error detection	Humidity in thermal insulation Short circuit of the wires Wire interruptions	Humidity in the indications (per welding point or pipe section) Short circuit of the wires Wire interruptions	Humidity in thermal insulation Short circuit of the wires Wire interruptions
Representation tolerance	1-3 % of the measuring distance (e.g. observation section) The sufficiently accurate localization of the leakage is carried out by iterative adjustment of the measuring distance.	< 1 m (per Indicator)	± 0.2 % of the measurement distance (e.g. observation section) The sufficiently accurate localization of the leakage is carried out by iterative adjustment of the measuring distance.
Control possibility	Central Decentral Manual	Central Decentral Manual	Central Decentral Manual
Simultaneous localization of several damages	It may be possible, if the loop is split	Possible, in case of division one error will be visible within the hierarchy	It may be possible, if the loop is split
Length of the observation section	Device related (up to 6.000 m loop length possible) Recommended Monitoring Section of 800 - 1.000 m Length Several loops can be connected	Hierarchical submission of the location section,4 hierarchies (0-3) 0. Hierarchy 1000 m 1-3. Hierarchy 1000 m Number of branches per hierarchy unlimited	Device related (up to 1.500 m loop length possible) Recommended monitoring section of 800 - 1.000 m Length Several loops can be connected

4.3.2.3 Documentation and testing

Documentation
For the optimal use of a monitoring and locating system, detailed and up-to-date documentation on the district heating network is absolutely necessary. Ideally, this also includes circuit and wiring diagrams, loop diagrams, information on the position and arrangement of sensors and other components as well as the approval protocol with information on tests and measurements. The various types of pipe used must be documented.

Testing
After connecting the leakage warning wires in the course of making the socket joints, the sensor loops must be checked for length, interruption, metallic contact and external voltages using an assembly testing device. An insulation test should be repeated about four weeks after commissioning the district heating pipe. The insulation resistance of the installed pipes should be in the mega-ohm range.

4.3.3 Fittings

Fittings are installed as shut-off valves. In this way, the interruption of network operation can be limited in the event of later network expansions and any necessary repairs. Fittings are also used for emptying and venting the pipes. The shut-off valves must meet the following requirements [5]:

- Minor pressure loss
- Impervious sealing in both directions
- Impermeability of the casing/ jackets
- Low maintenance requirements
- Low space requirement
- Low flow noise
- Interchangeability
- Thermal insulation possibility
- Durability of the casing / jacket
- Functionality even for infrequent use

Individual requirements influence each other, so that not all of them can be fulfilled at the same time and priorities must be set. Shut-off valves are not suitable for control purposes.

Valves can cause disturbing noises. VDI Guideline 3733 [91] contains a number of instructions for their suppression. When a valve is fully open, the noise development is usually negligible as long as certain flow velocities are not exceeded.

In district heating plants there are four basic types of shut-off fittings: gate valves, globe valves, stopcocks and butterfly valves. The casing can be manufactured in cast, forged or welded design. Depending on pressure and temperature conditions, the use of appropriately durable materials is essential. The valves are built into the pipeline either by welding or with flange connections.

4.3.3.1 Gate valve

The various design features result in the designations for gate valves shown in Table 4.4.

Table 4.4 Gate valve types [5].

Characteristic	Nomenclature
Shut-off element	Wedge gate valve
	Double knife gate valve
	Slide gate valve
Shape of the lid trim / vault	Flat slide
	Oval slide
	Round slide

The rigid wedge is not suitable for higher pressures and temperatures. Gate valves with elastic wedge or wedge plates can be used for all loads. Parallel gate valves are available with split and non-split end plates. Gate valves with non-split parallel sealing faces are not used in district heating pipelines. Split end plates with expanding wedges, which press the end plates onto the casing shortly before the closing process is completed, are more suitable.

Another important element is the valve stem which actuates the end plate. The types of spindle movement are shown in Table 4.6.

Materials for gate valves depend on the field of application. In the hot water range up to 110 °C and 16 bar lamellar grey cast iron can be used. For higher pressures and temperatures, materials with higher durability such as nodular cast iron, cast steel or forged steel should be used.

In most cases, the valves are affected not only by the internal pressure but also by forces and twisting resulting from the thermal expansion of the pipeline, therefore wear-resistant valve material is recommended. DIN 4757 requires that the casing of all valves in hot water pipes above DN 50 should be made of suitable materials even if cast iron or malleable cast iron would otherwise be permissible under the given operating conditions.

When using cast steel or welded constructions, the force absorption can be considerably increased. Gate valves up to about DN 250 are preferably hollow forged. Gate valves from about DN 50 upwards are also used in welded sheet steel constructions due to their durability. Gate valves should be rigid enough so that surrounding surfaces do not suffer any deformation due to pipe forces and pressure loads and that the tightness is not impaired.

Gate valves for district heating pipelines usually have the following design features:

- Flexible wedge or double plate wedge. Double plate wedge is resistant to deformation caused by forces and twisting from the pipeline. It is also easy to repair.
- External stem, if space permits. However, internal spindles are also common. Special attention must be paid to the stuffing box sealing. An internal spindle is only recommended from DN 200 upwards.
- Metallic sealing surfaces.
- Leakage rate 1 according to DIN 3230 part 3.
- Sufficient hardness and thickness of the casing on the sealing surfaces.
- Possibility of retrofitting an electric actuator.

4.3.3.2 Globe valve

A distinction is made between the valve types according to Table 4.5 for the present application range. In contrast to the gate valves, the valve seats do not slide on each other.

Table 4.5 Globe valve designs [5]:

Characteristic	Description
Seat type	Disc seat
	Cone seat
	Piston seat
Seat position	Straight seat
	Angle seat (inclined)
Lid design	Bonnet
	Headpiece
	Capless
Flow direction	Straight through
	Angle (corner)

Globe valves have the following advantages over gate valves, which justify their preferred use in the lower nominal size range:

- Simple design, thus lower manufacturing costs.
- Good sealing even at high temperatures and pressures.
- Easy adaptation by regrinding if the seat is damaged.
- The following criteria have a disadvantageous effect:
- Limitation of the nominal size due to the increased operating force towards the valve seat compared to gate valves and butterfly valves.
- Stronger flow diversion than gate valves and plug valves. They therefore have a higher pressure loss.

The valve bodies are manufactured in different shapes depending on the task. Parabolic plugs are required for certain control tasks.

Table 4.6 Classification of the gate valves according to the stem movement [5].

Criteria	Internal stem	External stem	
Rotary motion of the stem	Rotary motion	axial raising and rotary motion	axial raising
Motion of the hand wheel	non increasing	increasing	non increasing
Advantage	Low overall height with closed and open gate valve. No pollution of the stem thread.	Simple design. Better protection of the stem thread against the flowing medium. Lower actuation forces. Double lead of the stem. Visual inspection of the stem position. Hand wheel position corresponds to wedge position.	Simple design. Better protection of the stem thread against flowing medium. Lower actuation forces. Double lead of the stem. Visual inspection of the stem position. Stem position corresponds to wedge position. Minor space requirements.
Disadvantage	The stem thread is exposed to the flowing medium and is out of the range of visual control. Contamination by deposits is possible.	More space demand and inconvenient operation, due to the position of the hand wheel changes. Pollution on the thread is possible by external influences.	Manufacturing overhead, and therefore, more expensive. Pollution on the thread by external influences possible.

4.3.3.3 Plug and Ball Valves

Plug valves offer advantages due to their simple and robust design, small size, low pressure loss with favourable design options of the opening cross-section and fast switching times. The favourable flow avoids the accumulation of dirt. In taps, the sealing surfaces slide over each other during the switching process.

Dirty or damaged sealing surfaces can be repaired by simply grinding the seats or resealing. The valve cross-section usually corresponds to the pipe cross-section of the same nominal diameter.

For the operation of plug valves, a reduction gear in open or closed gear boxes is required from DN 200 upwards.

A typical representative of plug valves is the ball valve. Simplified, a ball valve consists of three parts, the ball plug, the globe seal and the body. The advantages are:

- Continuous flow in the pipeline, as long as the cross section is not constricted.
- Compact design.
- High stability against internal and external forces.
- No sealing agents.
- Low maintenance.
- High tightness with low sealing force.
- Protected seal position.

Body and ball of ball valves are usually made of steel or stainless steel. Up to PN 16, the body or parts thereof can also be made of nodular cast iron, and from PN 25 on also of cast steel.

Actuators can be connected via stem extensions to ensure favourable thermal insulation of the pipeline and the valve.

Valves with cylindrical plugs must be provided with a sealant. The valves must be resealed at certain intervals, regardless of the operating conditions and the frequency of operation. In addition to its sealing function, the sealant must also perform corrosion protection functions.

An O-ring seal made of Perbunan or PTFE is installed as a seal between the upper casing and the plug. A stuffing box packing is therefore not necessary. The sealant is pressed into the grooves of the plug and distributed over the sealing surfaces. Valves from DN 200 upwards have other filling openings in the valve body than the plug. To prevent the sealant from being pressed out by the medium, each opening is secured with a ball check valve. During sealing, the valve must be open and the plug must be moved repeatedly to achieve a good distribution of the sealant.

4.3.3.4 Flaps

Flaps or butterfly valves are shut-off devices whose closing bodies are predominantly in the form of a disc rotating on an axis within the cylindrical body. Due to their compact design, they have the advantage of requiring the least space compared to other shut-off devices and are also easy to operate. Flaps are partly designed so that they can be loaded from both sides.

The thickness of the circular disc depends on the operating pressure of the medium. If the rotary disc is outside the centre of the disc, interruptions of the sealing ring can be avoided.

The flap disc is adjusted via the drive shaft guided through the body on one side with an attached reduction gear. The position of the disc is divulged by a mechanically transmitted indicator.

The body is manufactured either with welding ends or connecting flanges. It can also be designed as a flangeless butterfly valve body for clamping between two pipeline flanges. Welded constructions for the body have proved to be very effective. When the seat ring of the disc seal is welded into the body, the disc carries the seal. Conversely, the profiled seal can also be arranged in the body.

In addition to pneumatic and hydraulic control cylinders, gearboxes are also available for the actuation of the butterfly valves and directly connected to the drive shaft of the valve disc. The gearboxes must be self-locking and designed according to the maximum torque of the butterfly valves, observing safety factors. For large actuator motors and when long shut-offs are required, spur gear reducers can be installed upstream. Worm gear, screw spindle and sliding crank gear are most frequently used.

Figure 4.5 shows the most common variants of mounting the valve disc in the body.

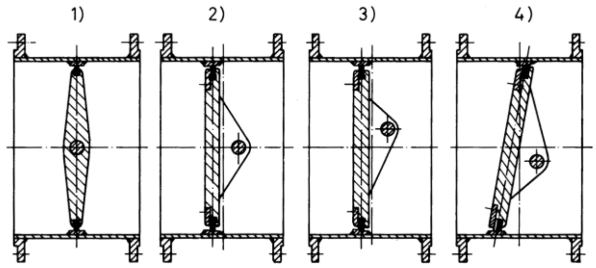


Figure 4.5 Arrangement of the valve discs [5]:
1 Centric bearing
2 Eccentric bearing
3 Double eccentric bearing
4 Double eccentric bearing (metallic sealing).

4.3.3.5 Area of application

Table 4.7 shows the range of application of the shut-off valves arranged according to nominal size ranges. The size is an important cost factor.

Because more than one of the valve types is usually suitable for the choice of a valve, Table 4.8 compares important valve properties

Table 4.7 Field of application of shut-off valves according to nominal widths [5].

Nominal Width	Type of armature
DN 40 -100	Main applications area are globe valves, plug and ball valves aswell as gate valves
DN 100 - 200	Within the overlapping range there are globe valves, gate valves, plug and ball valves aswell as butterfly valves
DN 200 - 350	Main applications area gate velves and butterfly valves aswell as plug and ball valves
> DN 400	Main applications area butterfly valves, gate velves aswell as plug and ball valves

Table 4.8 Qualitative comparison of the main properties of different valves [5].

Reference	Gate valve	Globe valve	Plug valve	Ball valve	Flap or Butterfly valve
Sealing surface	2 wide pairs of seal rings	1 small pair of seal rings	large seal area	large seal area	1 small seal ring
Overall length	small	high	high	medium	small
Overall height	large	medium	small	small	medium
Flow resistance	low	high	medium	small	high
Opening- / closing time	long	long	short	short	short
Operating frequency	few - medium	few	medium	frequently	medium
Eligibility for change in flow direction	good	not suitable	good	good	good
Range of application	medium until major DN	minor DN	minor until medium DN	minor until major DN	medium until major DN
Suitability for external forces and moments	Not sensitive Particularly suitable for the absorption of extended forces and moments are gate valves with round slides Most vulnerable are gate valves with flat sliders	Sensitive to high external loads	Specification of the load limit at which an actuation is still possible If this will be exceeded clamping and leaking is possible	Insensitive if without flange (e.g. welded)	Not very sensitive Suitable for the absorption of high forces without affecting the sealing
Reliability	Actuation necessary at least once a year	Actuation necessary at least once a year	Actuation necessary at least once a year	Actuation necessary at least once a year	Actuation necessary at least once a year
Life cycle	medium / long (40-60 Years)	medium (20-40 Years)	medium / long (40-60 Years)	long (60-80 Years)	medium / long (30-60 Years)
Costs of armature	low	low	medium	high	low (Wafer types valves) medium
Costs of armature incl. installation	medium high (for extended DN)	low	medium high (for extended DN)	medium low (for extended DN)	low (Wafer types valves) medium

4.3.3.6 Operational Information

Bypasses of the fittings are installed in district heating systems and transport lines. Bypasses are used to equalize pressure, prevent excessive pressure surges and make it easier to open the valves. DIN EN 12266 Part 1 must be observed with regard to the operation of valves [114]. Reference is also made to DVGW worksheet W332 [90].

Actuators
At high differential pressures and large nominal diameters high stem torques are required, which cannot be handled manually. The limit for gate valves and globe valves in the nominal size range around DN 200. The limit can be shifted upwards for gate valves by installing ball bearings. The torque can be reduced by mounting transmission gears.

In the following situations, operation by an electric drive is recommended:

- For large valves where manual operation is too strenuous.
- When manual operation cannot be carried out in the required time.
- In case of life-threatening danger.
- When a central control is provided.
- Automatic actuation due to impulse from pressure, temperature or level controllers.
- If specific procedures are scheduled.

Hydraulic actuators and occasionally pneumatic actuators can also be used for the remote control of valves.

External forces and torques
The absorption of forces and torques by the valve body is dealt with in DIN 3840.

If possible, valves should not be located in the area of high external loads and they should not be a fixed point in the pipeline. To relieve the load on the valves, constructive measures may be necessary on the pipeline. Reliable statements about the forces and torques to be expected at the installation site can be derived from a static or dynamic calculation.

The various valves react differently to external forces and torques. A classification of the valves is shown in Table 4.8.

Shut-down times and pressure surges
Every change in the flow rate causes pressure fluctuations in a pipeline, which propagate at the speed of sound (pressure surge or water hammer). Since such flow rate changes must be made frequently during operation, the repercussions of pressure fluctuations must be taken into account. Extreme load cases are decisive for the dimensioning of the pipeline. They can occur during emergency shut-downs and in case of sudden pump failure.

Experience shows that the shut-down times generally last a few minutes. The individual types of valves vary in shut-down and opening times. Fast closing and opening processes should be avoided as far as possible, even with nominal widths ≤ DN 150.

Functional safety
In order to ensure the function, it is recommended to operate the shut-off valves once a year.

4.3.3.7 Labeling and Documentation

The valves should be clearly and permanently labelled as follows:

- Nominal design pressure rating
- Maximum operating temperature
- Flow direction if necessary
- Manufacturer, type and serial number.

Actuators must be marked with labels:

- Direction of rotation of the output shaft
- Purpose
- Torque or drive power according to DIN
- Manufacturer, type and serial number.

The operator should have the following documents from the manufacturer at his disposal:

- Installation instructions
- User manuals
- Maintenance instructions
- Lists of spare parts and other fittings
- Product and material testing

4.4 Network structure

The term district heating network describes the link between heat generation and heat consumers. The choice of network structure, route, pipe system and installation method are influenced by numerous factors. Not only the regional layout but also technical, geological, economical and safety engineering, as well as architectural and legal criteria are important for the decision. The supply area and the size of a district heating network are therefore usually not fixed from the outset and can also change over time.

The district heating network is usually divided into main, branch and house connection pipes (Figure 4.6). The main pipe corresponds to the first pipe from the heat station. In the case of large heat generators located far away from the supply areas, the term transport pipeline is also used. Branch or distribution pipelines come off the main pipelines and serve in sub-distribution to the individual supply areas. The house connection pipes are used to connect a heat consumer to a main or branch pipe

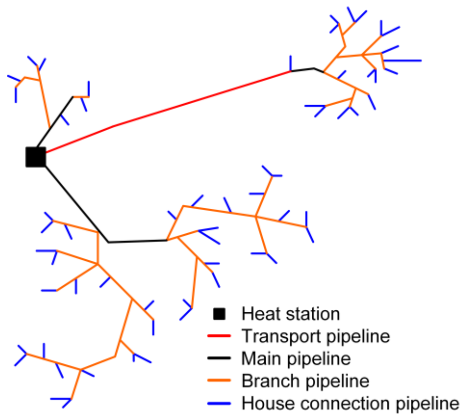


Figure 4.6 Network design and pipe types.

4.4.1 Subdivision by pipelines

Modern district heating networks are almost exclusively designed as closed **two-pipe systems** with water as the heat transfer medium and one supply and return pipe each (Figure 4.7). Special cases are the three-pipe and four-pipe systems [43].

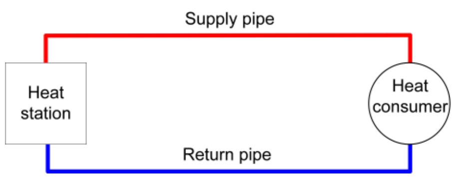


Figure 4.7 Two-pipe system.

The **three-pipe system** (Figure 4.8) consists of two supply and one return pipe or of one supply and two return pipes. With two supply pipes, for example, the customer's heating system runs continuously with one supply pipe, depending on the outside temperature, while the second supply pipe provides hot water at a constant supply temperature. In the case of two return lines, for example, the low return temperatures can be combined in one return pipe when using an exhaust gas condensation system.

Advantages of three-pipe systems are easier regulating and the lower heat loss due to the continuous supply. However, the third pipe makes the installation more expensive. In addition, the mass flow in the return pipe can fluctuate due to the discontinuous withdrawal from the second supply pipe shifting the operating point of the system. A further disadvantage is that during the transition period, the common return temperature can be higher than the temperature of the continuous supply, which can cause undesirable heating of the users´ rooms and increased energy consumption

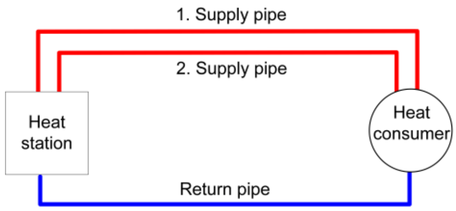


Figure 4.8 Three-pipe system.

A three-pipe system is also used for decentralized solar-thermal energy generation with seasonal storage. In this case, the two-pipe system is extended by a conductor that serves as the solar supply pipe for the central storage tank. The decentralised solar collectors are fed from the district heating return ([6], [43] and [56]).

The **four-pipe system** (Figure 4.9) consists of two separate two-pipe systems. For example, one can be operated with a constant supply temperature and one with a variable supply temperature. This improves the regulating possibilities and reduces heat losses. Since the four-pipe system is complex, it is only used if the two networks have different pressures, temperatures or different heating media.

With a decentralized solar thermal energy supply and central seasonal storage, there is also a four-pipe system, whereby a two-pipe system is used for district heating and another for the supply and return of the solar thermal system ([6], [43] and [56]).

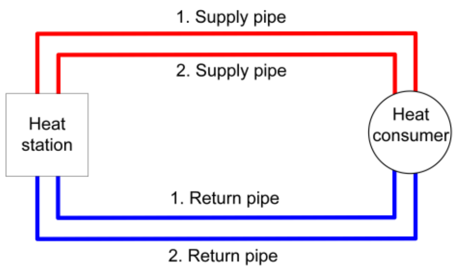


Figure 4.9 Four-pipe system.

4.4.2 Main distribution

The distribution network can be classified into two basic types, the star (tree) and the mesh networks [43].

Initially district heating networks are usually designed as star networks (Figure 4.10). Supply and return are usually of the same dimensions (symmetrical). The diameter of the pipelines is largest at the heat generator and decreases with distance from the heat generator. The pump delivery head is designed so that the agreed differential pressure is available at the end of the network at the last user. The short lengths of the pipelines and the small diameters result in low construction costs and heat losses. The disadvantage is that subsequent expansions are hydraulically problematic and the security of supply is lower, since the entire pipeline must be switched off in the event of a network failure.

A special case of the star network is the line network (Figure 4.11), which consists of only one main line with short house service pipes ([6], [43], and [56]).

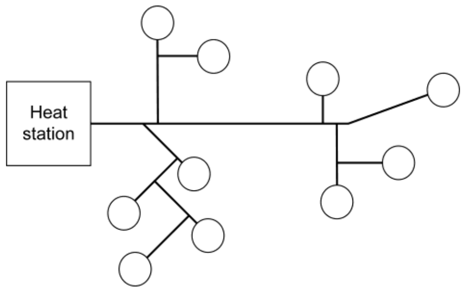


Figure 4.10 Star network with heat station.

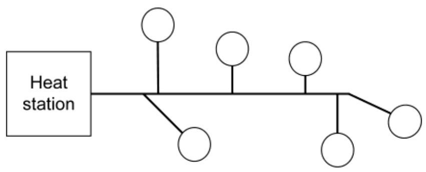


Figure 4.11 Line network with heat station.

As a result of growth and higher density of users as well as fusions of thermal networks, so-called mesh networks can be formed (Figure 4.12). They are characterized by a high reliability due to the multiple supply routes. In large urban supply areas, they are often the result of a continuous expansion of the district heating network. Mesh networks often have several heating centres and are operated in interconnected mode. However, pure mesh networks are not common as star networks are generally used to connect to the periphery.

A special case of the mesh network is the ring network (Figure 4.13), i.e. a supply system consisting of one ring with associated house connection pipes ([6], [43] and [56]).

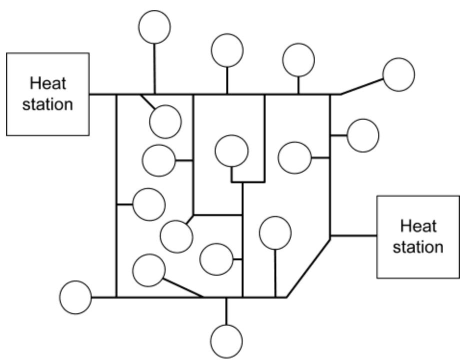


Figure 4.12 Mesh network with two heat stations.

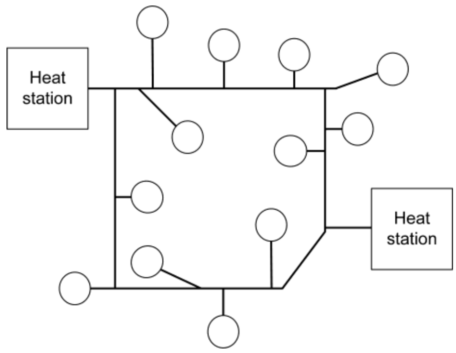


Figure 4.13 Ring network with two heat stations.

4.4.3 Sub-distribution and house connections

With a standard routing as shown in Figure 4.14, all consumers are connected separately to the main or branch pipeline. This frequently used routing via public roads offers the greatest flexibility in terms of connecting additional customers. However, low connection density or long house connection pipes may result in more pipework than with other routing systems. For this reason, as well as the branches and fittings required in densely built-up areas, high investment costs can result.

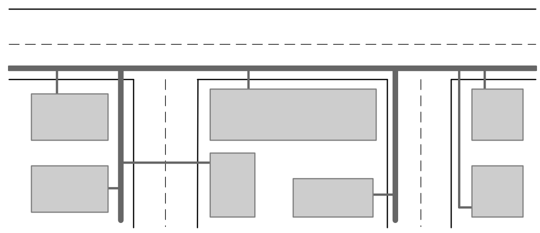


Figure 4.14 Standard-routing.

With house-to-house routing, houses are grouped together and only one house is connected to the main or branch pipeline. The other houses are connected from this one, so that fewer branches from the main or branch pipeline are necessary. Since the pipelines run through private property and buildings, the owners need rights of passage.

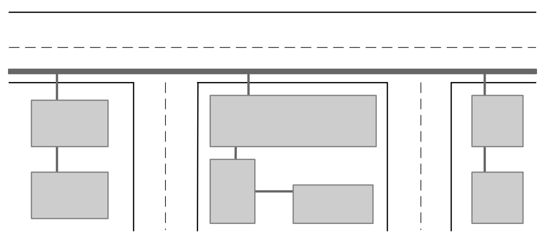


Figure 4.15 House-to-house-routing.

In the case of terraced houses, the best method for house-to-house pipe routing is installation in a cellar or basement. Laying pipes in a cellar is one of the most cost-effective installation methods. No civil engineering work is necessary nor do special district heating pipes have to be laid. Possible leaks are discovered quickly and can be located exactly. A prerequisite for laying in the cellar is that the houses or underground car parks are adjacent to each other and a short pipe connection between the individual heat substations is possible. In the case of buildings that are not adjacent to each other, a mixed form of house-to-house and cellar installation is also possible. The disadvantages are the more complex coordination of the individual construction phases and the coordination with the house owners for the connection rights.

A further, rarely used technique is the loop-in routing, which is not based on a network structure (star or mesh network), but rather closes all buildings by means of the door-to-door routing. This largely eliminates the need for buried pipe connections and branches. However, a later, unplanned network expansion is almost impossible. Therefore, the loop-in method is only advantageous for small district heating networks with flexible pipe systems.

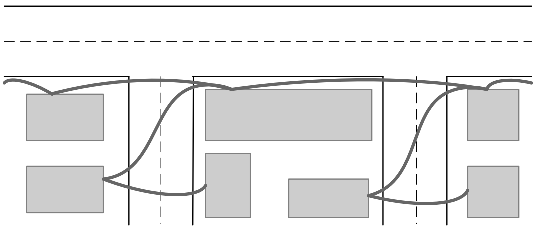


Figure 4.16 Loop-in-routing.

A combination of standard and house-to-house routing is usually cost-effective as it profits from the advantages of both systems.

4.4.4 Development of the network structure

The supply area and the connected load of district heating networks are often the result of a development process that can last for years or decades. Figure 4.17 shows typical stages in the development of a district heating network. The figures refer to characteristic phases and describe the following situations:

1) A single district heating network with a tree structure (see star network) is often built first. In many cases, smaller networks are also developed in other parts of the city, initially supplying heat to a specific group of similar buildings, zones or areas.
2) One or more of the smaller district heating networks are combined into a central network. A central heating plant can be integrated as a base load and external heating plants as peak load units.
3) Certain pairs of supply and return pipes are connected. Typically, these are the largest main strands, which then form a ring. With a ring network, more and more consumers can be connected.
4) The connection of further branches results in a mesh structure. The main pipelines usually follow the road structure, whereas this does not necessarily apply to branch and house connection lines.

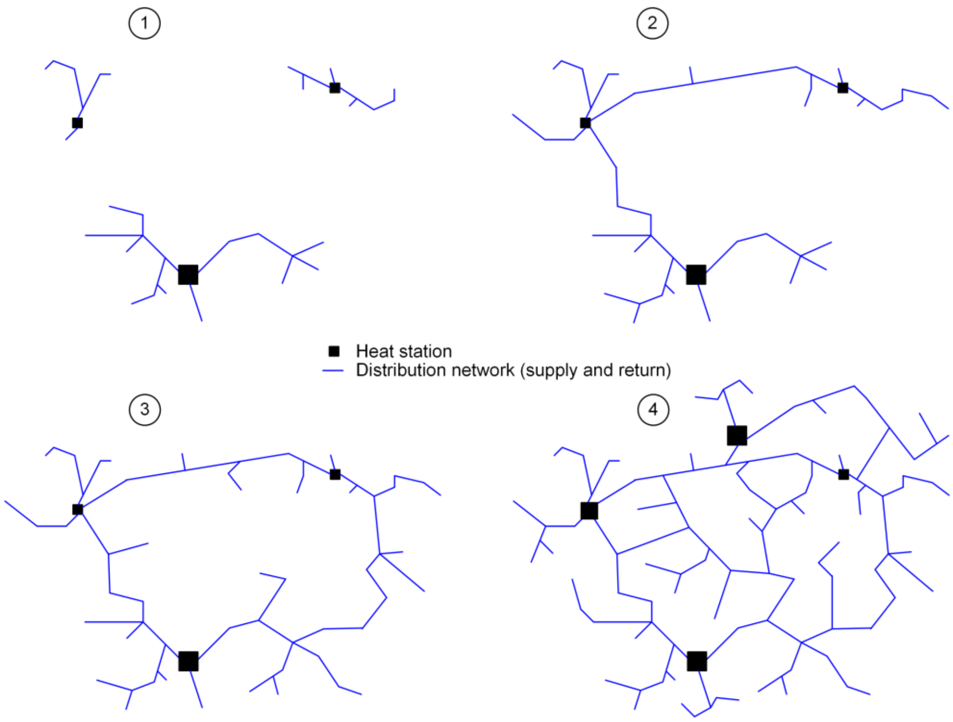


Figure 4.17 Four different network structures represent the growth process of district heating networks. 1) Star (Radial) network 2) Initial connection 3) Ring network 4) Meshed network.

4.4.5 Venting and draining

In the case of extensive district heating networks, it should be possible to empty the pipes in sections when branches and house connections are retrofitted or during repair work. Shut-off valves should be used to divide the district heating network so that individual sections can be emptied quickly and only limited quantities of treated water are needed to refill the pipes.

Valves are used as drain fittings. The designs according to chapter 4.4.3 also apply to them. It is recommended that the drains are provided with standardised hose connections. Drainage points should be designed as sludge traps.

If the district heating pipes are emptied into sewers, the normally permissible inlet temperature of 35 °C must not be exceeded. This can be achieved by adding cold water. During operation, the drains must be secured with blind flanges, plugs or caps.

To fill and empty the district heating pipes, air must be able to escape and flow in. For this purpose, vents and aerators must be installed at the highest points of the pipeline and between two shut-off valves. In the case of flat-laid branch pipes without significant gradients, the air can also be supplied and discharged via the house service pipe.

4.4.6 Measuring equipment

It must be possible to monitor the pressure conditions in the district heating network. Shut-off valves and pressure gauge shut-off valves with test connections for pressure gauges according to DIN 16271 [110] should therefore be installed at suitable points. They must be easily accessible. Pressure measuring points can be set up in drainage and ventilation pipes.

It is also recommended to install temperature measuring connections according to DIN 43772 [111] at suitable locations. For remote reading of the measured values it may be advisable to lay appropriate data cables when building the network.

4.5 Installation methods

4.5.1 Above ground installation

Above-ground installation is economically and operationally attractive. However, it requires extensive architectural and landscape design, for example for bridge crossing applications. In addition, weather conditions (UV radiation, frost, corrosion) or vandalism must be taken into account and appropriate measures taken.

The installation is preferably carried out on a support or suspended from hinged columns. The pipe supports are highly stressed by the fluid mass and the forces resulting from thermal expansion. Installation on supports and pipe bridges is usually limited to industrial sites and areas without special design requirements.

Figure 4.18 Above ground district heating pipes in Jena (Public Services Energie Jena-Pössneck).

4.5.2 Underground installation in ducts

Installation in inaccessible ducts is a solid construction with a long service life. Disadvantages are the high construction costs and the strict underground construction measures, which is why duct-laid routes are rarely used and will not be discussed further in this handbook.

4.5.3 Underground installation in trenches

Installation in trenches with pre-insulated jacket pipes has become the most common installation method because of its low costs and quick installation. The advantages are:

- Low expenditure on civil engineering work compared to other underground installation methods.
- Few costly shaft structures other than fittings that are not directly integrated into the jacket pipe system.
- Applicable even in difficult soil conditions for example in areas with groundwater.
- Short assembly times due to a high degree of prefabricated components.
- Great flexibility in the routing through prefabricated fittings.

Particular care is required in the manufacturing and thermal insulation of the connections at the construction site, which is why this work is done as a service by most system providers or by specialised companies.

Figure 4.19 District heating pipes will be lifted into the trench [51].

4.5.4 Trenchless installation

Installation of pipes underneath objects (roads, railway lines, watercourses) are realised by means of digging techniques such as flush boring, press driving, culverts and tunnelling. With trenchless installation techniques, a distinction is made between soil displacement and soil extraction methods. The latter are divided into controllable and non-controllable methods. In the case of the non-controllable methods, a sufficiently precise alignment of the bore hole must be carried out at the beginning. As a rule, the medium-carrying pipes are drawn in through a protective pipe.

Highly specialised equipment is required for pipe jacking.

4.5.4.1 Soil displacement methods

A displacement cone driven into the ground creates a cavity into which the pipe is pushed. The method is used up to DN 200. The force is applied either by a pneumatically driven displacement hammer in the ground or by a horizontal ram in the starting shaft.

4.5.4.2 Soil extraction methods

The soil is loosened by a rotating drill head or a horizontally rammed open pipe and mechanically, hydraulically or pneumatically transported into the starting pit (flush drilling method, Figure 4.20). Plastic or steel pipes can be drawn through the resulting tube.

In the jacking method, a hydraulic press is used to press pipes up to 4 m in diameter from the starting pit to lengths of up to 150 m to the target pit. At the starting and target points, shafts are required to accommodate the drilling equipment and to insert and install the pipes.

Drilling over long distances is also possible with a remote-controlled drill head or by means of a flush drilling method in loose sediments and semi-solid sediments with a maximum grain size of 63 mm. The drilled material is flushed out by the water at the drill heads or by a clay-containing suspension (bentonite). When the drill head is extended backwards, the pipeline is pulled into the compacted drill channel. For stony subsoil, the construction of a concrete micro-tunnel with a remote-controlled tunnelling machine may be used. The micro-tunnel is then flooded with water, allowing the pipe to be used as a floating body.

Figure 4.20 Horizontal flush drilling.

These methods enable pipelines to be laid with high precision under roads, railway embankments, rivers or buildings. Changes of direction are carried out with wide radii if possible. Pipe bursting techniques are commonly used for the renovation of existing pipes, in which the old pipes are destroyed and displaced into the ground and new pipes are laid

4.5.4.3 Culverts

Culverts are structures used to pass under roads, a tunnel, rivers or railway tracks. Water bodies are crossed mostly by culverts. The crossing of narrow, small waters is relatively simple. Often the elasticity of the pipeline is sufficient to guide it through the water, whereby anchoring and covering is necessary. If this is not possible, a pipe trench (culvert channel) is created in the water bed (Figure 4.21). The pre-assembled pipe is lifted into the culvert trench using mobile cranes, taking precautions to protect the thermal insulation, and is loaded for buoyancy control.

Large culverts for waters are prefabricated on land. The same regulations for bending, welding, corrosion protection and pressure testing apply as for buried pipes. The prefabricated culvert is laid in the culvert channel according to the following procedures:

- Pre-flush procedure: For flexible pipes with small diameters the culvert is settled into the water bed using vibration.
- Lowering procedure: The prefabricated culvert is lowered into the culvert channel by cranes onshore or from a ship.
- Tow-in method: The culvert is pre-assembled on land on sledges which stand on a skid track. The culvert is then pulled into the prepared culvert channel from the opposite bank of the river (Figure 4.21).

The culvert is encased in concrete or fibre cement to protect it against mechanical damage. The jacket also serves as a buoyancy protection. Open culvert channel laying is being replaced by the Directionally Controlled Horizontal Drilling method. This method is economical, but requires exact preparation and installation.

Figure 4.21A prefabricated culvert is lifted into the culvert channel placed into the riverbed on rolls [51].

4.6 Frequent installation situations

The following explanations describe frequent installation situations with advantages and disadvantages for the different pipe systems. It should be noted that not all pipe systems are suitable for installation, depending on pressure, temperature and pipe diameter.

4.6.1 Fixed surfaces

In urban district heating networks, laying along public roads and paths is preferred. The district heating companies acquire the right to lay in the public road area or conclude concession contracts with the municipalities [6]. Attempts are made to lay the most important main pipes along less busy roads (green areas, footpaths), which are easily accessible in the event of a fault. Road crossings should be reduced to a minimum [6]. Earth-laid pre-insulated rigid and flexible steel pipes and pre-insulated PEX pipes have become the preferred choice for laying along roads or on unpaved surfaces. Duct installation methods are now used only rarely and in special cases. The use of pre-insulated jacket pipes (pre-insulated rigid steel, flexible steel and PEX pipes) is optimal due to their standardisation, robustness, small space requirement and low installation costs [6].

For situations without crossing pipes or other obstacles, routing is usually unproblematic. For pre-insulated rigid steel pipes there is the option of welding the pipes into larger sections and laying them with the so-called pipeline installation [43]. Flexible pipe systems, on the other hand, have the advantage that they can be laid directly off the roll and are also self-compensating [43]. For situations with crossing pipes, routing with rigid pipe systems can be complex and for pre-insulated rigid steel pipes require the use of additional fittings such as 90° elbows [43].

4.6.2 Unpaved surfaces

For pipes under unpaved surfaces (e.g. cultivated land), the route can be largely optimised to suit the selected pipe system and implemented cost-effectively.

4.6.3 Routing on sections of terrain

It is often possible to lay a district heating pipeline and other supply lines along a road, a railway line or a watercourse, which is sometimes already taken into account in zone planning. This use requires an agreement with the respective owner [6].

If rivers, motorways or railway tracks have to be crossed over or under, the route can be accommodated in a culvert or in a special third-party structure such as a railway bridge. However, the use of third-party structures carries the risk that considerable adaptation costs can incur if changes are made. Trenchless installation methods have proven to be effective for underpasses [6].

4.6.4 Private property

For every pipe that is laid on private property, its use must be legally secured. This can be done by legal registration or by agreement between the network operator and the owner [6].

4.6.5 Consumers to be connected subsequently

If it is known during the construction of the district heating network that new buildings are planned at a later date or existing consumers are to be connected, preparation is necessary. The corresponding house connection pipes can, for example, already be routed into the garden and equipped with a shut-off valve or a steel end piece with the possibility for drilling [43].

In the case of pre-insulated rigid steel pipes, an unplanned subsequent connection can be made by drilling the distribution pipe during operation. With pre-insulated PEX pipes, a T-piece can be retrofitted by freezing, squeezing or temporarily shutting down the system.

4.6.6 Consideration of other pipeline related applications

The descriptions within chapter 4.6.6 relate to the “AGFW” publication «Bau von Fernwärmenetzen» [5]. They describe the consideration of other pipeline related service lines. A sufficient minimum distance must be maintained from the supply lines (gas, water, electricity) available in the road space. In addition to the valid guidelines, local regulations must be considered. Furthermore, general basic recommendations are listed below.

4.6.6.1 Power and telecommunication cables

The transmission capacity of power cables laid in the ground decreases with rising ground temperature [5]. The minimum clearances between district heating lines and power and telecommunication cables are recommended in Table 4.9.

When running power cables and district heating lines in parallel, adequate space must be allowed for installation work on the cables. Particular attention must be paid to shafts that exceed the normal profile of the district heating pipe [5].

Direct current (DC) railway line systems or other DC systems may pose a risk of corrosion for underground district heating lines due to their stray currents [5].

Table 4.9 Minimum clearance between district heating pipes and power and telecommunications cables at crossings and parallel routing [5].

Crossing and parallel routing up to 5 m length	Minimum distance [cm]
1 kV-signal- or measurement wire	30
10 kV-Wire or a 30 kV cable	60
Several 30 kV-cables or cables above 60 kV	100
Parallel routing with >5 m length	**Minimum distance [cm]**
1 kV-signal- or measurement wire	30
10 kV-cable or a 30 kV cable	70
Several 30 kV-cables or cables above 60 kV	150

4.6.6.2 Gas and water pipes

If district heating pipes are laid parallel to gas or water pipes, there should be a minimum clearance of 40 cm between the district heating pipe and the external pipe. The greatest possible distance from gas pipes must be maintained, as defective gas pipes can pose considerable risks. Gas pipes may not cross district heating ducts and shafts without protection. Even when crossing with other pipes, a distance of 20 to 30 cm should be maintained [5].

Table 4.10 Minimum clear distance between district heating pipes and gas and water pipes at crossings and parallel routing[5].

Crossing	Minimum distance [cm]
Gas line	As large as possible; however ≥ 30
Water line	20 – 30
Parallel routing	**Minimum distance [cm]**
Gas line	As large as possible; however ≥ 40
Water line	≥ 40

If pipes must lay in concrete channels, jacket pipes should be used or other constructive measures should be considered [5].

4.6.6.3 Sewage and rainwater pipes

Sewer pipes are usually deeper than district heating pipes. Nevertheless, district heating pipes should not be laid parallel over sewers, so that repair work is always possible without damage to the pipes. Laying over sewer pipes is possible only if this sewer can be repaired from the inside. [5].

If one pipe is built over another, the static conditions must be observed [5].

When planning the district heating route, underground structures such as access shafts, storage chambers and retention basins must be taken into account. Manhole covers must be opened to determine the position of the underground structure. Drainage pipes to the sewer can often be an obstacle. During planning, it may be more cost efficient to relocate the district heating pipe [5].

4.7 Civil engineering

The civil engineering work in district heating pipeline construction involves digging trenches for underground pipes and the subsequent repair of the surface. To be considered are ducts, shaft structures for ventilation, emptying, cable pulling, sectioning, pumping stations, network separation and heat transfer stations, as well as trenchless laying methods or tunnelling when passing under objects. This chapter provides an overview of civil engineering techniques, standards, guidelines and further literature for civil engineering design. It describes the procedures and requirements for the appropriate design of district heating systems. The focus here is on the methods of direct laying that are common today. Special methods such as laying in concrete ducts or shafts are not dealt with in detail. Especially in densely populated areas, careful planning is absolutely necessary.

4.7.1 General information

No district heating network can be built without civil engineering works. In fact, the rules of civil engineering have a decisive influence on the pipe routing and the depth of the pipes. Road surroundings, for example, are usually already occupied by plant pipelines, and district heating must make do with the remaining space available. Buildings cannot be accessed by the shortest route, as busy roads or freshly renovated areas prevent this. The planner must be familiar with these constraints and take them into account accordingly.

Of the total investment and operating costs of a heating network, the largest share, usually around 60 %, is for the pipe network [15]. Of this, 60% is for the costs of pipeline construction (engineering and pipes). Of these pipeline construction costs, around 60 % is spent on civil engineering work. Careful planning of the pipeline layout therefore helps to improve the profitability of a project. For example, a route along main roads should be avoided.

One of the basics for route planning is the survey of all existing pipelines. Some municipal utilities also have their own work specifications which must be taken into account in the planning process. The assessment of the pipe statics is also important for the pipe routing. For rigid pipe systems, especially with dimensions > DN 80, it is advisable to create longitudinal profiles in built-up areas.

In addition, it is advisable to take into account construction site logistics and traffic routing during the construction phase as early as possible and to contact the responsible authorities of municipalities, states or the federal government.

In order to ensure a long service life of buried district heating pipes, a high standard of execution is required.

4.7.2 Route Planning

4.7.2.1 Rigid pipe systems

In principle, when planning the route, care must be taken to ensure that the pipes run to the users over the shortest possible distances. After determining the horizontal position, the depth must be defined. It must be taken into account that, in addition to the main pipe, house connections must also be made, which ideally should branch upwards. Experience has shown that, due to the fact that roads are often already heavily occupied with factory pipes, main pipes should be laid at the same depth as existing water pipes (see also section 4.6).

When the direction of a pipe deviates from its course, great attention must be paid to pipe statics (chapter 7.5). Direction deviation occurs in the following cases:

- changes in inclination
- obstacles
- changes in direction
- pipe jacking

Figure 4.22 until Figure 4.25 show recommendations on how to plan changes of direction.

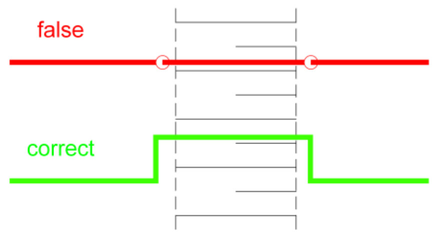


Figure 4.22 Changes in inclination

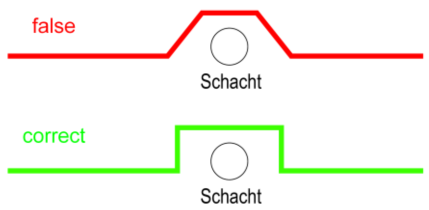


Figure 4.23 Obstacle

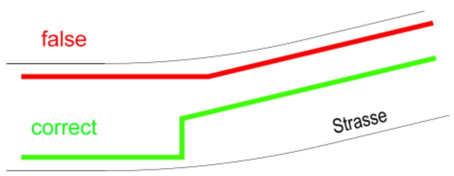


Figure 4.24 Changes in direction (elbow or elastic bending as an alternative)

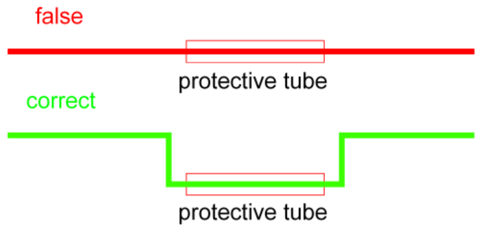


Figure 4.25 Pipe jacking

In order to optimise the routing when changing direction, it is also possible to bend straight composite pipes by cold forming and to use so-called curved pipes (Figure 4.24). When bending, care must be taken to ensure that no unacceptable stresses and deformations (ovalisations) occur in the steel service pipe, in the insulation, in the jacket pipe and in the wires of the leakage monitoring system. Elbow pipes can be manufactured in the factory or on site using appropriate bending tools. The jacket pipe must be protected against damage. Minimum bending radii for curved pipes bent on site can be found in the AGFW data sheet FW401 part 9. Further notes on the steel construction are found in part 10 [100].

Venting of elevated points is mandatory and it should be possible to shut off entire pipes with shut-off valves. District heating pipes larger than DN 100 should also be drainable. Shafts made of cast concrete or prefabricated products are used. Manhole structures are also used for shut-off valves and cable ducts for control systems and leak monitoring. Many heat supply companies already have regulations on how such shafts are to be designed. Alternatively, and at a much lower cost, the pipe can be blown through with compressed air via venting and draining fittings.

Due to the fact that pre-insulated rigid steel pipes are generally large diameter systems, route deviations are sometimes necessary, obstacles can compromise pipe statics (e.g. electric pipe blocks), etc., it is recommended to use longitudinal profiles. Below is a list of advantages of creating a longitudinal profile.

- Planning reliability
- Identification of venting and draining sites
- Basis for civil engineering tendering (standard position catalogue [78] distinguishes between different trench depths)
- Detection of critical depth levels (too much or too little covering of pipes)
- Identification of conflict points with existing service pipelines
- Identification of laying depth for static analysis
- The relationship between terrain and basement elevation level is established for house connections.

The design of the trenches depends on the pipe dimensions, the trench depths and the terrain (road or grassland). It should be noted that in expansion zones a larger clearance trench width is required for mounting. A distinction is made between trench shoring and sloped trenches.

Basically, trench shoring is mainly used near roads whereas sloped trenches are used in grassland. The type of trench is to be selected according to Federal Construction Work Regulations [77] and according to DIN 4124 [109].

4.7.2.2 Flexible pipe systems

Planning the route for flexible pipes is much easier than for rigid pipes. Except for any transition between rigid and flexible pipes, there are usually no pipe static limitations to be considered. When using pre-insulated plastic pipes, the pipe statics are completely negligible.

Flexible pipe systems are used extensively, usually up to DN 100, but the main area of application is for smaller pipe dimensions such as house connections. Larger dimensions are used, for example, for overcoming steep slopes or for crossing streams.

When planning, it is particularly important to ensure that the radii are sufficiently large for installation. Particularly with larger dimensions, the minimum radii specified by the manufacturers are often insufficient which is illustrated in the example below (Figure 4.26).

Figure 4.26 Radii for the laying of flexible pipe sytems.

A major advantage of flexible pipes is quick installation, as the trench is narrow and can be constructed without grouting. The laying of the pipes normally takes one day, joints are installed the second day.

4.7.2.3 Obtaining permits

A further point in route planning concerns obtaining the necessary permits. For example, it may make economic sense to lay a district heating supply pipeline in undeveloped private land. However, in such a case, servitude must be clarified before taking further steps, such as applying for building permits or tendering.

Preliminary clarifications must also be made with the authorities as to how to obtain the necessary permits. The differences between the counties and states can be considerable, also with regard to the duration of the procedure. For example, in some states a simple application for excavation is sufficient, while in others applying for a building permit is necessary.

It must be clarified whether roads may be crossed or used and how they must be repaired. It is also advisable to conduct a survey of other utility companies that could also benefit from the planned construction. This cooperation can lead to lower construction costs for district heating

4.7.3 Construction procedure

The construction work for district heating pipelines is basically divided into 4 phases:

- Trench excavation (incl. wall ducts and shaft constructions)
- Pipeline construction
- Trench backfilling
- Surface repairs

The civil engineering contractor may face interruptions of several days or even months in the case of repairs. It is up to the planner to draw up a time frame that allows the civil engineering contractor to work as uninterruptedly as possible.

For pipe construction, the longest possible building lots are desirable. The main advantages of this are fewer pressure tests, the possibility of pre-stressing, avoiding unnecessary joints in flexible pipes and minor installations. All of the points mentioned have an impact on construction costs and total construction time.

Prior to the start of construction, an inspection must be carried out by the site management, in which the civil engineering contractor, the pipe fitter and possibly the representatives of the authorities responsible for traffic must take part. In addition to the installation, intermediate pipe storage areas, pipe lifting openings and the power and water supply must be clarified. The fact sheets of the AGFW FW 401, part 12 and 18 [100] are also to be observed.

4.7.3.1 Trench excavation

If the building company or heat supplier provide no specifications, the specifications of the pipe suppliers apply for trench excavation. In Switzerland the SUVA [87] and [86] regulations, the BauAV [77] and DIN 4124 [109] always take precedence over these specifications. The clearance of trench width can be derived from the following pictures.

For compact and stable material (e.g. clay), trench shoring is not necessary up to a trench depth of 1.50 m (Figure 4.27). For a trench depth of over 1.50 m, trenches must be shored or built with sloping sides for safety reasons. Due to long and often simultaneous trench building phases, large amounts of bracing material are needed which the civil engineering company must provide.

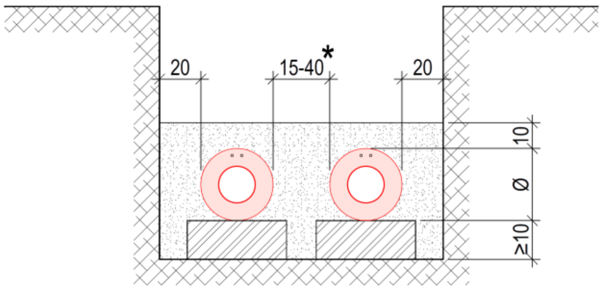


Figure 4.27 Clearance of trench width without sloping (* according to manufacturer's information; specification in cm)

The X:Y ratio for sloped trenches (Figure 4.28) depends on the excavated material. For example, the ratio of slope inclination in solid material such as clay is 1:3, while in loose material such as gravel it is 1:1.

For pipe laying it is not relevant if **excavated material** can be stored laterally or on a construction site deposit or must be removed. However, laid pipes are covered by specially provided material, therefore a considerable amount of excavated material must be removed.

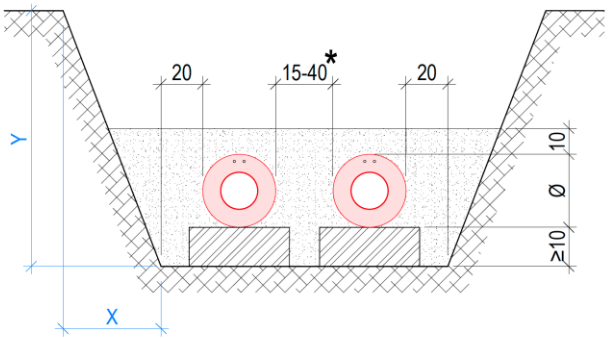


Figure 4.28 Clearance of trench width with sloping (* according to manufacturer's information; specification in cm).

Exposed **service pipelines** must be secured and protected in accordance with the requirements of the pipeline contractors.

In the case of flexible pipe systems, the trench is filled with the pipe coating material to a thickness of at least 10 cm before the pipe is laid.

Wall ducts are generally installed for house connections, when supply pipes run through underground parking areas or for shaft structures made of cast concrete. Connections must be water and gas tight. In order to meet these requirements, it is imperative that no lateral, axial or radial movements on the part of the pipes affect the connection. The pipe routing must be planned accordingly.

Ducts are normally made by core drilling. It should be clarified in advance how the ducts are to be sealed. Possible sealing methods include sealing rings made of neoprene rubber, which are pushed over the pipes, or ring space seals. Table 4.11 shows the different dimensions of the core drillings. The decisive factor is the diameter of the jacket pipe (e.g. the PE jacket pipe for pre-insulated rigid steel pipes).

Table 4.11 Diameter of core drilling in mm depending on pipe dimensions and different seal types.

Diameter core drilling	Diameter of the jacket of the pre-insulated rigid steel pipe											
	110	125	140	160	180	200	225	250	280	315	355	400
Wall sealing ring	200	200	250	250	300	300	350	350	400	400	450	500
Annular space sealing	200	200	200	250	250	300	300	350	350	400	450	500

The area of application of the simpler wall sealing rings is in areas with well-permeable soils with an inclination towards the wall ducts. In all other areas, as well as on slopes, in waterlogged areas or with a gradient towards the wall duct, annular space seals must be used. It should be noted that when using the annular space seals, they must not be plastered inside the buildings, as they must be tightened if necessary. In new buildings, so-called multi-section house connections (Figure 4.29) with the independent sealing of individual pipes (water, district heating, electricity, etc.) are also suitable.

Figure 4.29 Example: Multi-Sector house connection- Source: DOYMA GmbH.

For dimensioning and execution, use the manufacturer's instructions and AGFW FW 401 [100] Part 7. The requirements for wall ducts are laid down in DIN 18195 [107] and should be observed accordingly.

A distinction is made between **accessible and non-accessible shafts**. Due to their size, accessible shafts are mainly constructed in cast concrete. The minimum requirements of worksheet FW 433 of the AGFW [103] as well as the SUVA regulations ([87] and [86]) and the BauAV [77] apply to the construction. These regulations include the following points:

- Minimum height of the interior of shafts for operating valves (1.8 m) and minimum area for operating work (1.5 m^2), minimum width (1 m), service corridors (min. 500 mm)
- Shaft ventilation (natural circulation)
- Shaft bottom with pump sump or floor drain
- Shaft entrance (min. 0.8 m, resp. 0.6 m) and equipment

In the case of the non-accessible shafts (e.g. with well rings), special attention must be paid to their foundation, especially in the road area.

4.7.3.2 Pipeline construction

In this phase the civil engineering work is paused. Here it may be necessary to seek assistance for pipe insertion, shoring or the creation of prestressing fixed points.

Chapters 4.4 to 4.6 deal with the network construction, different installation methods and typical installation situations. However, this handbook does not include more detailed documentation on pipe installation, such as the execution of welding work, joint installation, leak test, etc.

Further literature on pipeline construction can be found, for example, in the rules and regulations and other publications by the Arbeitsgemeinschaft für Wärme und Heizkraftwirtschaft AGFW e.V. such as the Technical Handbook for District Heating [6] or the Installation Handbook [8] by the Bundesverband Fernwärmeleitungen e.V. (Federal Association of District Heating Pipelines.

In principle, only companies that can prove their technical expertise should be commissioned to lay pipes. In Switzerland, it is common practice, and usually mandatory in public tenders, that the company responsible for pipeline construction must offer references. Usually the reference for pipeline construction includes two similar plants of comparable size. Emplyed welders must have the required welding certificates (welding class B). In Germany and Austria, public tenders also require the company to be certified in accordance with the AGFW worksheet FW 601 [106].

After completion of the welding work, the welding seams must be inspected. As a rule, a visual inspection (according to DIN EN ISO 17637) is followed by a non-destructive weld seam inspection to the correspondingly specified extent. In the case of a radiographic test, the test class B of DIN EN 1435 should be aimed for. A liquid penetrant test shall be performed in accordance with DIN EN 571-1, an ultrasonic test in accordance with EN 1714, a magnetic particle test in accordance with DIN EN ISO 17638 and an eddy current test in accordance with DIN 54141.

After the non-destructive test, the leak tightness and/or strength test is carried out in accordance with AGFW-Merk-blatt FW 602. The visual methods with air are recommended as a standard test rather than with water. Weld seams are covered with a foam-forming agent. If no formation of bubbles is detected within one minute, the tightness is considered to be proven. For the method with internal air overpressure, the test pressure is 0.2 to 0.5 bar, with external air under pressure (vacuum goggles) a maximum of 0.6 bar absolute. A cold-water pressure test on the ventilated route must be carried out in accordance with DVGW Code of Practice G 469, Procedure A1. The test pressure is 1.3 times the operating pressure at the high point and must be maintained for 3 hours.

4.7.3.3 Trench backfilling

After completion of the pipeline construction (with socket and expansion pad installation as the last step), the route must be approved by a responsible site manager before the sand bed can be created.

The pipe covering with fine-grained material consists of silt sand (grain size 0-0.75 mm) or washed round gravel (grain size 0-4 mm, no crushed materials allowed!). In the case of rigid systems, care must be taken to ensure that the pipes are neatly tamped. Special attention must be paid to the spaces between the pipes and any cavities. These spaces must be filled and compressed separately. This will prevent later settling and displacements. Any auxiliary supports must be removed simultaneously, except for sandbags that are to be slit open or hard foam supports. This work step must always be carried out by hand. Slurring of the pipe coating is not permitted! The pipes must be surrounded by at least 10 cm of this fine-grained material. It is recommended that a route warning tape be laid on the pipe covering for each pipe.

If washing out of the bedding sand during civil engineering work cannot be ruled out, e.g. by rain, the bedding zone must be covered with geotextiles. This should be especially observed on slopes or steep sections because of the drainage effect of the trench profile. When in contact with water, sand does not always satisfy the degree of compaction (D_{Pr} x 97%). The grain sizes are segregated so that the nominal friction coefficients cannot be achieved on the pre-insulated rigid steel pipes and the so-called 'tunnel effect' is produced. Therefore, according to AGFW FW 401 part 12, slurrying of sand is not classified as state of the art.

If free-flowing bedding materials such as self-stabilising sand mixtures (SSM) or floor mortars are used, it should be noted that there is no long-term experience with the removal of these with simple equipment. In practice, there are also no permanent and reliable test results available for the mechanical characteristics such as long-term friction behaviour. A general approval of these backfill materials as road construction material by the Forschungsgesellschaft für Straßen- und Verkehrswesen FGSV (Reasearch Association of Roads and Transport), has not yet been granted. These are not taken into account in the pipe static principles according to AGFW FW 401 part 10 and 11.

Replacement materials such as foam glass granulates, crushed sand, recycled material or similar are generally not permitted in the pipe zone as bedding material or sand bed.

Protective measures may be necessary if the covering pipe material is above or below the permissible height. If the maximum cover height is exceeded according to the pipe supplier's specifications, the use of prefabricated concrete U-profiles is recommended. If the height of 60 cm is not reached in the road area, load distribution plates can be installed above the pipe covering to reduce wheel pressure (Figure 4.30).

In grassland, especially in gardens where covering is less than 65 cm the laying of concrete panels is generally recommended (Figure 4.31). This measure serves as a protection against mechanical damage (e.g. by driving in poles).

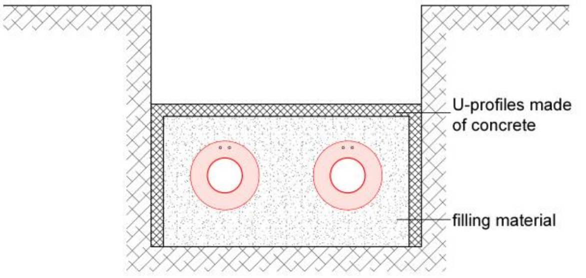


Figure 4.30 Example of protection measures if the maximum coverage height is exceeded with load distribution plates made of prefabricated concrete U-profiles.

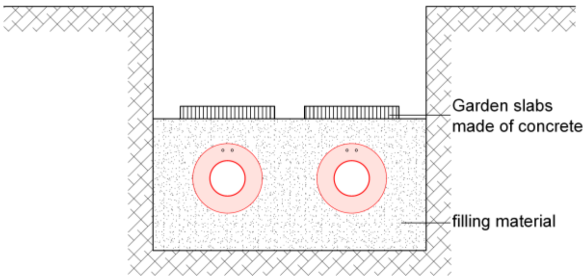


Figure 4.31 Sample protection against mechanical damage with garden slabs of concrete.

After covering the pipes, the actual trench backfilling takes place with excavated or compactable material (e.g. gravel mix 0/45, frost-proof). The material is to be placed in layers of approx. 30 to 50 cm thickness each and compacted. Any exposed utility pipelines must be reinforced with lean concrete. Any trench obstructions can be removed in this phase. Depending on the depth of the trench, it is possible to lay cable protection pipes for the control system or leakage monitoring on the first intermediate level. Great importance must be attached to compacting, since improperly executed work can cause damage later on, especially on roads.

The uppermost part of the trench consists of topsoil (grassland) or the surface including the pavement layer (roads, pavements). The construction of covering surfaces must be carried out in accordance with regional regulations or SN 640535C [92].

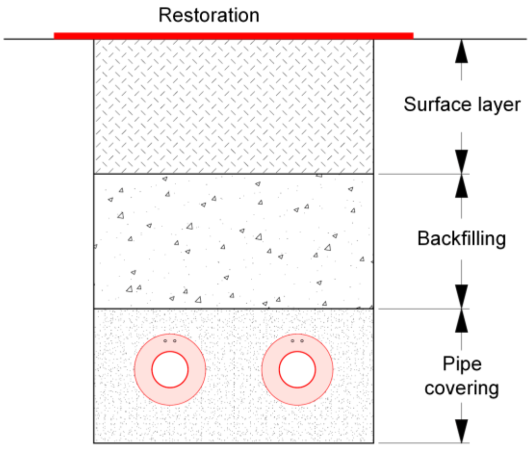


Figure 4.32 Schematic diagram (layering) of the trench fill.

4.7.3.4 Restoration

Repairing the surface and bringing it back to its original condition before construction work is referred to as restoration. In cultivated land the implementation is by far the easiest to accomplish. It is more complex in gardens, where lawns have to be resown, bushes replanted or pavements replaced. In the road sector, the local authorities (road managers) usually determine the extent of the (surface) pavement to be repaired.

Restoration is often carried out after the actual civil engineering work. Reasons for this are e.g. replanting of gardens, which is carried out by gardening companies and not by the civil engineering contractor. The resurfacing of roads is not carried out during the cold months.

4.8 Water quality

In order to avoid damage caused by corrosion, erosion or overstressing of materials in the system, the district heating water must meet certain requirements that also depend on the heat generation. A comprehensive description is provided by the AGFW regulations FW 510 [6] and the SWKI guideline BT102-01 [91]. The following explanations summarize the most important points.

4.8.1 Water types

In this chapter, the terms commonly used in building and district heating technology for the different types of water are briefly described. Special attention is paid to the term hot water, which is commonly used in building and district heating technology. For this reason, in this document, the terms domestic hot water and domestic hot water generation are used for hot drinking water.

The fluid categories defined according to SVGW are also listed [88]. According to these liquid categories, the circuit water in the district heating network (primary and secondary) usually corresponds to categories 3 and 4.

4.8.1.1 Terms used in building technology

Drinking water:
According to Swiss food legislation, drinking water is defined as water that is left in its natural state or after treatment for drinking, cooking, food preparation and cleaning of objects that come into contact with food [67]. Drinking water also includes water for personal hygiene and washing (shower water, bath water, etc.).

Cold water:
Cold drinking water whose temperature is not deliberately raised.

Domestic Hot water:
Medium and high temperature hot drinking water is defined as water whose temperature is increased by the application of heat, not to be confused with hot water in district heating technology.

For this reason, the terms domestic hot water and domestic hot water generation will be used.

Operating Water
Water for commercial and domestic use, which does not have to be of drinking water quality.

4.8.1.2 Terms used in district heating

Untreated water:
The water available before the treatment plant, regardless of any previous off-site treatment.

Soft water:
Water freed from alkaline earths by ion exchange (dehardened).

Desalinated water:
Water largely freed from dissociated water-soluble substances by ion exchange, characterised by an electrical conductivity < 20 µS/cm and a silicic acid content < 0.5 mg/l.

Fully demineralized water:
Water freed from dissociated water-soluble substances by ion exchange (also known as deionised, demineralised or salt-free water), characterised by an electrical conductivity < 0.2 µS/cm and a silicic acid content < 0.02 mg/l.

Fill and make-up water:
Treated water used for filling, refilling and partial filling of district heating networks.

Circuit water:
Water that flows in district heating networks through the heat generator, the district heating network and possibly through heating elements or heat transfer stations. The term applies not only to primary networks but also to water in a secondary network.

Hot water:
Circulating water with a temperature ≤ 110 °C. Does not necessarily have to be of drinking water quality and should not be confused with hot water in building technology.

High temperature water:
Circulating water with a temperature > 110 °C.

4.8.1.3 Liquid categories

Liquids that may come into contact with drinking water are divided into five categories [88]:

Category one:
Water for human consumption, which is taken directly from a drinking water installation and complies with food legislation regulations.

Category 2:
Liquid that does not pose a risk to human health. Liquids suitable for human consumption, including water from a drinking water installation, which may have a change in taste, smell, colour or temperature (heating or cooling).

Example: Hot drinking water, fruit juices, soups, coffee, water from cooked food, etc.

Category 3:
Liquid which presents a hazard to humans due to the presence of one or more toxic substances.

Example: Heating water (without additives), water with antifreeze, rinse water for dishes and kitchen appliances, cistern water, dental workplaces, etc.

Category 4:
Liquid which presents a hazard to humans due to the presence of one or more toxic or particularly toxic substances or one or more radioactive, mutagenic or carcinogenic substances.

Example: Heating water with additives, water with surface-active substances, disinfectants, algicides, etc. The difference between category 3 and 4 is LD50 = 200 mg/kg body weight.

Category 5:
Liquid that poses a health hazard to humans due to the presence of microbial or viral agents of transmissible diseases. Examples: Rainwater, swimming pool water, washing machine water, toilet water, water from animal watering places, etc.

4.8.2 Recommendation on water quality in district heating technology

The recommendations given in this chapter concern the circuit water in the district heating network (primary and secondary side) and serve to prevent water-bearing damage due to corrosion and deposits and to ensure high energy efficiency. The circuit water in the district heating network usually corresponds to fluid category 3 or 4. The applications are differentiated according to the operating temperature and size of the system. If the quality of the raw water deviates from the standard values, corrective measures (water preparation, water treatment) must be taken. Further information on measures, methods of water preparation and water treatment can be found in chapter 12.2 of the appendix.

4.8.2.1 Hot water up to 110 °C

Table 4.12 shows guide values for filling and supplementary water and in

Table 4.13 for circuit water. In addition to these guide values, the requirements of the component manufacturers must also be taken into account.

An annual check of the circuit water is recommended.

Table 4.12 Requirements for the fill-up and make up water up to 110 °C [91].

Explanation	Unit	Reference
Electrical conductivity	µS/cm	< 100
pH-Value	–	6.0-8.5
Overall hardness	mmol/l	< 0.1

Table 4.13 Requirements for the circulating water up to 110 °C [91].

Explanation	Unit	Reference
Electrical conductivity	µS/cm	< 200
pH-Value	–	8.2-10
Overall hardness	mmol/l	< 0.5
Chloride	mg/l	< 30
Sulfate	mg/l	< 50
Oxygen	mg/l	< 0.1
Dissolved iron	mg/l	< 0.5
Total organic carbon-content	mg/l	< 30

4.8.2.2 High temperature water above 110 °C

In district heating networks, two water-chemical modes of operation are common for circulation water temperatures above 110 °C:

- - low-saline operation
- - saline operation

Table 4.14 lists guide values for the fill-up and make-up water and Table 4.15 for the circuit water from 110 °C. In addition to these guide values, requirements of the component manufacturers must also be taken into account.

A quarterly check of the circuit water is recommended.

Table 4.14 Requirements for the fill-up and makeup water from 110 °C [91].

Explanation	Unit	Water chemical operation set points		
		Low-salt		salty
General		colorless, clear, free from non-dissolved particles		
Electrical conductivity	µS/cm	10 - 30	>30 - 100	100 - 1500
pH-value		8.0-10	8-10.5	8.5-10.5
Overall hardness	mmol/l	< 0.02	< 0.02	< 0.02
Oxygen	mg/l	< 0.1	< 0.1	< 0.1

Table 4.15 Requirements for circulating water from 110 °C [91].

Explanation	Unit	Water chemical operating mode set points		
		Low salt		salty
General		colorless, clear, free of undissolved constituents		
Electrical conductivity	µS/cm	10 - 30	>30 - 100	100 - 1500
pH-Value		9.0-10.0	9.0-10.5	9.5-10.5
Total hardness	mmol/l	< 0.02	< 0.02	< 0.02
Oxygen	mg/l	< 0.1	< 0.05	< 0.02
$K_{S8.2}$ (p-Value)*	mmol/l	–	0.1-0.5	0.5-5
Phosphate	mg/l	3-6	5-10	5-15

The acid capacity $K_{S8,2}$ indicates how much acid a water sample absorbs to the transition point of the indicator phenolphthalein (pH 8.2). This measurement provides conclusive information about the lye concentration present in the water and is given in mg/l. This measurement should also be carried out in the steam boiler area.

5 Heat Transfer – Basics

This chapter describes the connection of the domestic heat distribution to a district heating network. In principle, any building can be connected to a district heating network, provided that a suitable building system for heating and, if necessary, hot water heating or ventilation is available or is being installed. The type, scope and technical details for a connection to a heating network are specified in a heat supply contract.

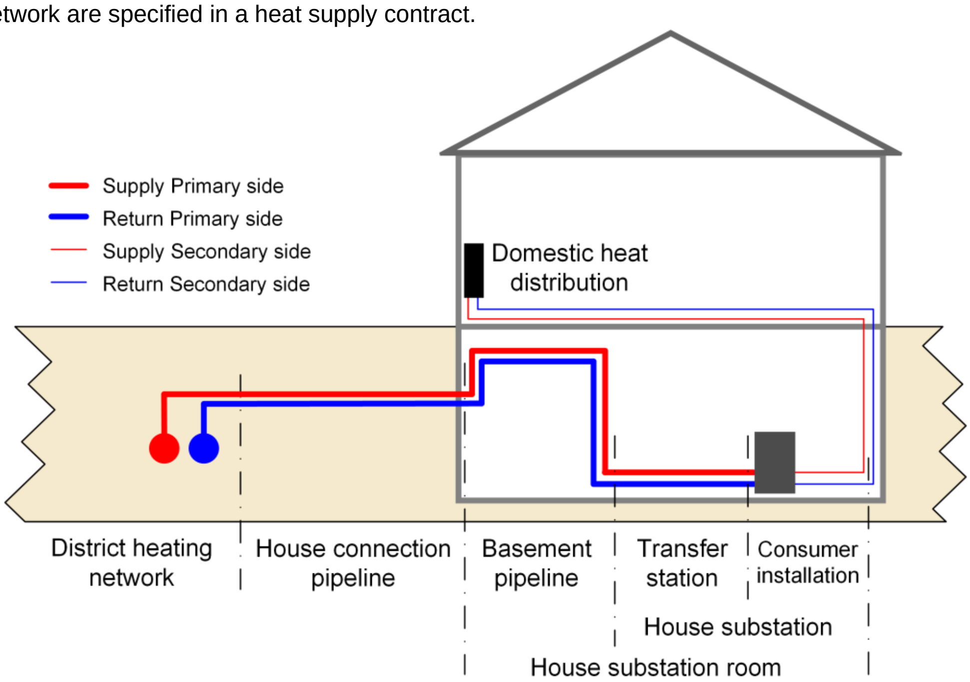


Figure 5.1 Overview of the house connection.

5.1 Terms and definitions

Figure 5.1 shows the most important terms for heat transfer based on DIN 4747 [115], VDI 2036 [116] and the explanations in [6] and [56].

5.1.1 House connection pipe

The house connection pipe connects the district heating network to the transfer station and is usually part of the operating facilities of the heat supplier. Design and execution are carried out by the heat supplier. The routing to the transfer station must be agreed between the parties involved. The house connection pipe from the wall duct into the building to the transfer station is also called basement pipeline and is equipped with a shut-off valve.

5.1.2 House substation room

The necessary connection facilities and operating equipment are installed in a house substation room. Location and dimensions are to be coordinated with the heat supplier at an early stage. The basis for planning is in DIN 18012 [112]. A house substation room is not required for a detached or semi-detached house. The following requirements must be observed when designing the house substation room:

- - The room should be located close to the point of entry of the house connection pipe.
- - The room should be lockable and always accessible.
- - Due to noise emission, the room should not be located near noise-sensitive rooms such as bedrooms.
- - The room should be adequately ventilated.
- - Sufficient lighting is required for maintenance and repair work.
- - An electrical connection is required for the heat substation.
- - The room should have an adequate drainage system so that, if necessary, the system can be emptied during maintenance and repair work. A fresh-water tap should also be available.

5.1.3 Heat transfer station

DIN 4747 [115] defines the heat transfer station as the link between the house connection pipe and the consumer installation. The heat transfer station (also called transfer station) is used to transfer the heat or the heat transfer medium to the consumer installation in accordance with the regulations regarding pressure, temperature and volume flow.

In the meantime, it has become common practice not to regard the heat transfer station as an independent unit, but to combine it with the function of the consumer installation which is sometimes called a compact station. Nevertheless, the transfer station traditionally represents the boundary of ownership between heat supplier and heat consumer.

The conditions for the design of the transfer station are set by the heat supplier with the technical connection regulations (TCR) which are part of the heat supply contract. Possible combinations between heat supplier and consumer installation must also be agreed upon, which is particularly essential for compact stations. However, individual functions are often used by both partners, e.g. control valves that secure the customer's system with an emergency control function.

Chapter 8.1 deals with the hydraulics and technology of transfer stations and the necessary equipment.

5.1.4 Consumer installation

The consumer installation is the link between the substation and the domestic heat distribution. It serves to adjust the heat supply to the domestic heat distribution with regard to pressure, temperature and volume flow.

When designing the central house unit, a distinction must be made between direct (Figure 5.2) or indirect connection (Figure 5.3). With indirect connection, the heating medium of the heat consumer is separated from the heat carrier of the heat supplier by a heat exchanger. In this case, the design of the domestic heat distribution (distribution pipes in the building, heating surfaces) can be designed independently. The connection of the domestic hot water heating system also offers various possibilities such as storage tanks, an instantaneous domestic hot water preperation and storage tank charging with or without priority function.

5.1.5 House substation

The house substation consists of the transfer station and the consumer installation and can be designed for direct or indirect connection. The type of connection is usually specified by the heat supplier. In the past, the boundary of ownership and responsibility between the heat supplier and the heat consumer was often in the house station, usually between the transfer station and the consumer installation. In practice, these installations were assembled separately and also procured, installed, operated and maintained by the respective entitled party. This is still the case with large plants.

For smaller and medium-sized systems, compact stations are used almost exclusively today, in which the transfer station and the consumer installation are combined into one structural unit. Due to the construction method and different operator models, the boundaries of ownership and responsibility within a heat substation are flexible. However, the same technical functions as in separate units can still be found.

Whenever possible, a data connection to the heat station should be integrated into the heat substation. This simplifies invoicing, fault recording and system optimisation

5.1.6 Domestic heat distribution

A sufficient heat supply requires coordination between heat supply and application. For this reason, the heat supplier needs information about the domestic heat distribution system. As the last link in the chain and interface to the user, the domestic heat distribution system has the task of making the heat available in the desired form. It is usually owned by the building proprietor who is responsible for its installation, operation and maintenance.

5.2 Connection options

In addition to the network operating modes, network types and operating parameters, the type of connection determines the design of the heat substation. The requirements for the design of the district heating network connection are specified by the heat supplier in the technical connection regulations which are part of the heat supply contract. A basic distinction is made between direct and indirect house connections.

The control or regulation of the house connections by individual controllers is the simplest solution for smaller systems. For medium-sized and larger systems, a solution via a ecntralized control system should be considered. With this central monitoring option, both errors in the parameterization of the individual controllers and optimisation possibilities can be identified. By recording data from the individual heat meters, it is no longer necessary to make readings on site.

5.2.1 Direct connection

With a direct connection, the heating water from the district heating network flows through the domestic heat distribution system (Figure 5.2). This has the following consequences for the construction and operation of the consumer installation and the domestic heat distribution system:

- Water quality is set and monitored by the heat supplier. However, water quality must be taken into account when choosing the material for the consumer installation and domestic heat distribution system to prevent corrosion.
- A pressure control system is not necessary, as the required system pressure is guaranteed by the heat supplier. If the maximum network supply pressure falls below the permissible level, pressure safeguarding is not necessary. Otherwise, a pressure reduction safety device must be provided.
- A return flow addition easily enables an exact adaptation of the operating mode to the requirements of the domestic heat distribution.
- A maximum network supply temperature below the permissible supply temperature of the domestic heat distribution eases the burden on the system. Neither a return flow addition nor temperature safeguarding are necessary.

A direct connection is only possible if the static pressure and the return pressure of the district heating network are lower than the permissible pressure of the domestic heat distribution.

5.2.2 Indirect connection

With indirect connection, the heating water from the district heating network does not flow through the domestic heat distribution system, but is hydraulically separated from the heating medium by a heat exchanger in the heat transfer station (Figure 5.3). This is necessary if the parameters in the district heating network (pressure, temperature, water quality) are not suitable for domestic heat distribution and it has the following consequences:

- With indirect connection, there are two heating circuits. The part through which the district heating water flows is referred to as the primary circuit (primary side), the part through which the heating medium of the domestic heat distribution system flows is referred to as the secondary circuit (secondary side).
- The pressure level and the pressure rating of the system components of the domestic system can be determined freely. This makes it possible to design the domestic installation with a lower pressure level making the system more economical, which in many cases justifies the somewhat more complex heat substation.
- The pressure and temperature safeguarding depends on the domestic heat distribution system. Therefore, even old systems with dubious pressure levels can be connected without any major risk.
- The circulation of the district heating water and the compensation of the temperature-induced volume change must take place in the heat transfer tation.
- The choice of materials and connecting elements on the secondary side is independent of the water quality in the district heating network.
- When designing the primary-side plant components, the maximum possible pressures and temperatures of the district heating network must be taken into account.
-

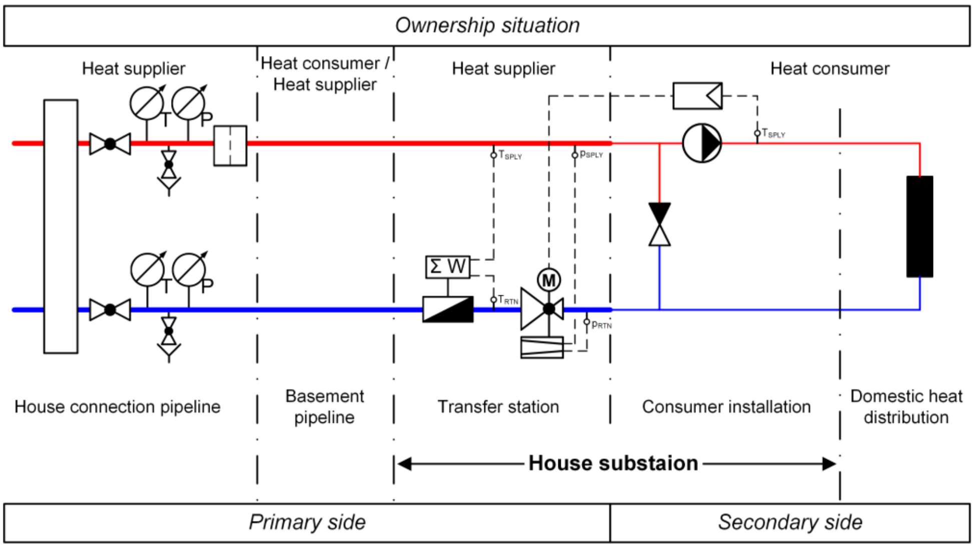


Figure 5.2 House connection with direct heat transfer.

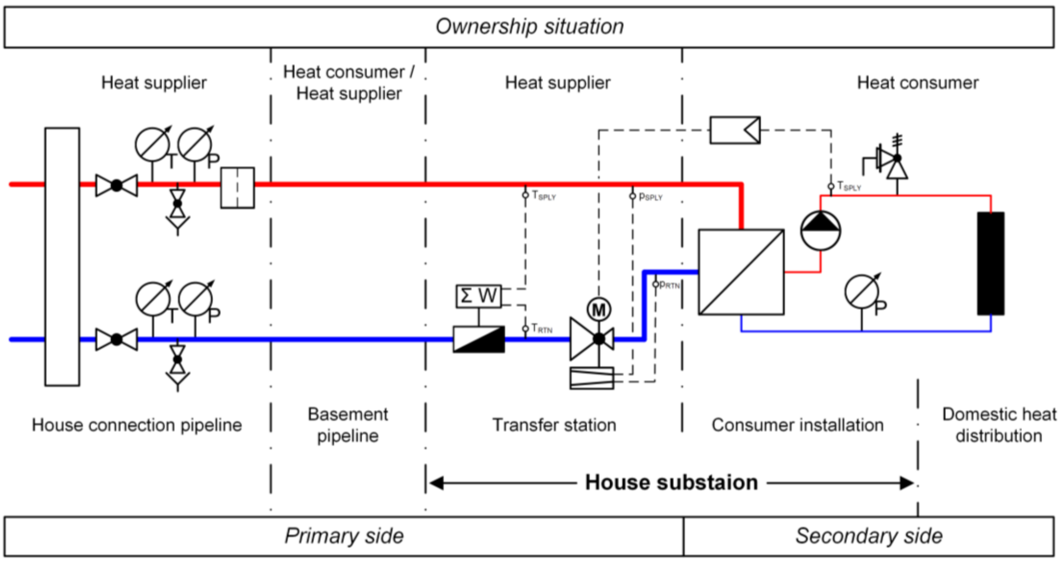


Figure 5.3 House connection with indirect heat transfer.

5.3 Heat supply contract

A heat supply contract establishes a long-term partnership between the heat supplier and the heat consumer. Most issues involved are legal and technical in nature and can have considerable economic consequences for both parties. It is therefore recommended to consult appropriate consultants or to use model contracts. There are many possibilities in the drafting process and a great variety of existing heat supply contracts. The following points need to be clarified between the contracting parties:

- What the heat price includes and an estimation of values to be expected
- Contents of technical connection requirements and connected load of the heat consumer.
- Scope and type of heat supply, usage and ownership
- Redundancy / security of supply and procedure in case of faults in the system
- Planned contract term, termination conditions, procedure for changes of ownership
- Legal framework for energy production and routing
- Funding opportunities

Various heat supply model contracts are available including those of Holzenergie Schweiz [55] and CARMEN [52], which serve as a basis for the following design descriptions. The points below must be defined and agreed in the heat supply contract:

- Parties
- Contract components and order of priority
- Purpose
- Term of contract
- Connection to the heat supply network
    - Construction, operation, maintenance
    - Interfaces, ownership
    - Connected load
- Prices (usually excluding VAT)
- Meter readings, payments, due date
- Troubleshooting service
- Final clauses

The heat supply contract should be supplemented with the following documents, if they are not already included in the contract:

- General terms and conditions of the heat supply contract (GTC)
- Technical connection regulations (TCR)
- Tariffs

5.3.1 General Terms and Conditions (GTC)

The General Terms and Conditions (GTC) serve as the basis for a large number of contracts and define the following points:

- Terms
- Construction, operation, maintenance and ownership
- Obligation to supply and purchase heat
- Prevention of supply interruptions - Liability of the heat supplier
- Damage mitigation
- Heat transfer control by third parties
- Rights of transit, access and use
- Change of the connected load
- Termination of heat supply
- Liability of the heat consumer
- Change of ownership
- Procedure in case of incorrect readings
- Premature termination of the contract
- Contract amendments

5.3.2 Technical Connection Requirements (TCR)

The technical connection requirements (TCR) are an integral part of the heat supply contract and serve the heat supplier and the heat consumer as a guideline for ensuring the technical aspects of planning, implementation and operation. The TCR apply to the following points:

- Scope of application and terms
- Heat transfer medium
- Sealing
- Pressures and temperatures
- Heat exchanger
- Domestic hot water heating
- Heat transfer station and boiler room
- Hydraulic integration and control
- Materials / Connections
- Temperature and volume flow rate limits
- Assembly, inspection and commissioning
- Maintenance

Further information on the TCR is given in the AGFW leaflet FW 515 - Technical connection conditions - heating water [105].

The following diagrams are recommended as supplements to the TCR:

- District heating supply and return temperatures (primary side) as a function of the outside temperature
- Hydraulic diagram of house substation
- Hydraulic diagram of domestic hot water heating.

5.3.3 Tariffs

A list of tariffs is part of the heat supply contract and refers to the costs involved in the heat supply contract. The heat supply agreement covers several factors relating to the price of heat. It makes sense to divide the costs into consumption-dependent and consumption-independent costs. The costs of investment, maintenance and servicing are largely independent of consumption. Consumption-related costs cover fuel and electricity consumption. The price for district heating often consists of three components:

- One-off **connection fee** for the construction and commissioning of the house connection. The payment is usually made after completion of the house connection.
- **Basic price** per kW connection capacity per year to cover fixed costs.
- **Energy price** per kWh of heat supplied.

The following points are usually regulated in the list of tariffs:

- Preliminary remark
- Tariff system
- Prices / price adjustment clauses
  - Connection fee
  - Basic price
  - Energy price
- Special connection conditions.

In heat supply contracts, the adjustment of cost components is often regulated by indices that reflect the price development of, for example, energy prices (for example, for wood or heating oil as a basis for comparison) and are relevant for the heat production costs. Recommendations and examples of price adjustment clauses are manifold. C.A.R.M.E.N.e.V. has published helpful hints on this topic in its leaflet on price adjustment clauses and price indices [53] and the Arbeitsgemeinschaft für Wärme und Heizkraftwirtschaft AGFW in its guideline on price calculation and changes in district heating prices [54].

Depending on the type of heat generation, the heat production costs can be significantly influenced by the temperature level of the heat. For example, the energy consumption of heat pumps, CHP systems and wood-fired heating plants with exhaust gas condensation increases considerably due to higher supply temperatures and even more so due to higher return temperatures. For this reason, it is conceivable that in future contractual temperature levels will be taken into account for the remuneration.

One possible approach is that not the consumed energy but the volume is taken into account during a billing period. This would create an incentive for end users to plan and install an efficient domestic heating system or, in the event of renovation, to make appropriate changes (low return temperatures result in a correspondingly low volume).

Planning and Calculation

6 Project Schedule

6.1 Overview

The overall system of a district heating network comprises heat generation, heat distribution and heat delivery to the customers. The following description of the project schedule includes the planning tasks for the district heating network, i.e. heat distribution from heat generation to heat delivery to the customers. Figure 6.1 shows an overview of the project schedule. A clear distinction is made between the **design phase** and the **operating phase**.

The **design phase** is subdivided in four phases:

Phase 1: Preliminary study

Phase 2: Conceptual design

Phase 3: Planning, tendering and awarding of contracts

Phase 4: Execution and approval

The planning is described by four **checklists**:

Table 6.10: Checklist Phase 1

Table 6.11: Checklist Phase 2

Table 6.12: Checklist Phase 3

Table 6.13: Checklist Phase 4

After the design phase and/or after approval, the district heating developer formally takes over operational management, supported by the planner if necessary in specific cases and for operational optimisation.

The **operating phase** comprises the following two phases:

Phase 5: Operational optimisation

Phase 6: Operation and management

The operating phase will be described by two **Checklists:**

Table 6.14: Checklist Phase 5

Table 6.15: Checklist Phase 6

6.2 Quality Assurance

When designing, executing and operating district heating networks, it must be ensured that the agreed quality is maintained during planning, construction and installation. This is done by defining the quality and monitoring execution based on the requirements. Quality assurance is not limited to individual phases, but is a process that takes place across all project phases.

AGFW “Worksheet FW 401 Part 17” [100] sets out the necessary quality assurance measures for building a district heating network with pre-insulated rigid steel pipes. The strategies presented in it can also be applied to other pipe systems [6].

The above-mentioned worksheet is closely aligned with SN EN ISO 9001 [95]. It assumes a holistic approach to the integration of customer requirements and customer satisfaction. Therefore, the AGFW worksheet FW 401 Part 17 should be understood as a recommendation for action aligned with SN EN ISO 9001.

Depending on the situation, execution according to SN EN ISO 9001 and applying the checklists is generally required (e.g. public sector) and in all other cases strongly recommended.

In Worksheet FW 401 Part 17, AGFW has created checklists that provide an overview of which quality assurance measures must be carried out at which points in the construction project. These checklists are intended for customers or their representatives as initiators of construction projects, and contractors such as civil engineering firms or pipeline constructors.

Table 6.1 Extract from the checklists for the customer and the contractor divided into execution sections according to [100].

Section	Verification Activities
Project preparation	Designing and selecting a suitable installation concept, completeness of the execution documents, tendering, monitoring execution, etc.
Civil engineering	Contractor’s compliance with traffic safety rules, pipe trench construction, handover to pipeline manufacturer, surface restoration, etc.
Pipeline construction	Inspecting the pipelines and other materials, pipe laying and fitting, checking the quality of welding seams, tightness,
Sleeve mounting	Validating the credentials of specialist personnel, controlling sleeves and expansion pads, approving pipeline construction and sleeve fitting,
Project completion	Joint approval of the work (client and contractor), completeness of the documentation, handover to the customer.

Table 6.2 Planning procedures and regulations in force in Switzerland, Germany and Austria and a project schedule optimised for this handbook. The regulations use different terminology and there are also differences in the respective planning stages.

Switzerland **SIA-Order 108 [80]** **Issue 2014** **Phases and sub phases**	**Germany** **HOAI [62]** **4th revised edition 2013** **work phases**	**Austria** **General Terms and Conditions of the Technical Offices - Engineering Offices of Austria [63]**	**Handbook on Planning of District Heating Networks** **Project Phases**
1 Strategic Planning 11 Formulation of needs, solution strategies	1. Basic determination		**(Guideline VFS)**
2 Preliminary studies 21 Project definition, feasibility study 22 Selection process	2. Preliminary planning (project and planning preparation)	1. Preliminary planning (project and planning preparation)	**1. Preliminary study**
3 Project planning 31 Preliminary project	3. Conceptual design (system integration planning)	2. Conceptual design (system integration planning)	**2. Conceptual design**
32 Construction project 33 Approval procedure, planning project 4 Tender 41 Tender, tender comparison, tender application (preparation of tender documents)	4. Approval planning 5. Execution planning 6. Preparation of the award 7. Involvement in the awarding of contracts	3. Approval planning (submission planning) 4. Execution planning 5. Preparation of the award 6. Involvement in the awarding of contracts	**3. Planning, tendering and awarding of contracts**
5 Realization 51 Implementation project 52 Execution 53 Commissioning, completion	8. Site supervision (construction supervision)	7. Site Supervision (construction supervision), acceptance, invoice verification 8. Acceptance	**4. Execution and approval**
6 Management 61 Operation, operational optimisation 62 Preservation	9. Property support and documentation	9. Audit	**5. Operational optimisation** **6. Operation and management**

6.3 Differences between SIA 108 and this handbook

Table 6.2 shows the terms used in the planning processes for various sets of regulations in Switzerland, Germany and Austria and the use of these terms in this Handbook on Planning of District Heating Networks.

The main differences between SIA 108 and the planning procedure in this handbook are as follows:

- Strategic planning (SIA 108 Phase 1) is defined in the guideline of the Swiss District Heating Association (VFS). The Handbook on Planning of District Heating Networks essentially starts with phase 2 according to SIA 108 (phase 2 preliminary study).
- In the Handbook on Planning of District Heating Networks, the construction project and tender phase according to SIA 108 (phases 3 and 4) is combined in one phase (phase 3 planning, tendering and awarding).
- The management phase (SIA 108 Phase 6) is divided into two phases in the Handbook on Planning of District Heating Networks. Phase 5 is dedicated to the management and phase 6 to operation and management.

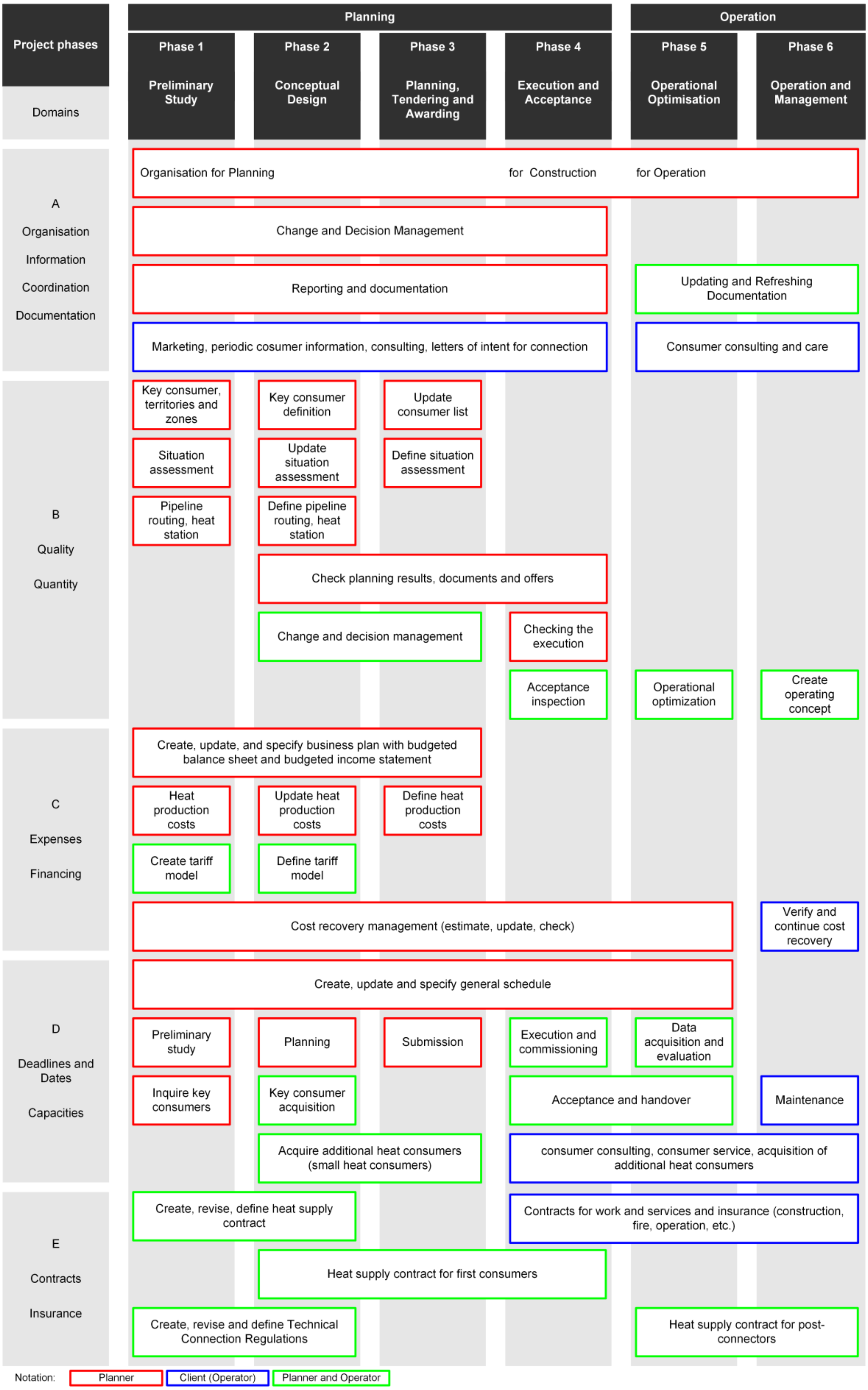


Figure 6.1 Overview of the Project.

6.4 Phase 1: Preliminary study

The aim of the preliminary study is to make binding statements on feasibility, risks and benefits. Clearly agreed goals and a **situation assessment** form the basis for this.

The aim of the situation assessment is to create the design and decision-making basis for a technically and economically optimal district heating network. In particular, it is important to find the optimal network structure for possible individual expansion stages and finally for the final capacity stage in an iterative process. On the basis of this situation analysis, the heat distribution and the heat transfer can be optimally designed. Depending on the progress of the project, the following points must be recorded:

1. Heat supply area
2. For every customer space heating, domestic hot water, process heat:
   a) Heat demand
   b) Heat load demand
   c) Required temperature level
3. Annual heat demand of the entire plant
4. Load characteristic curve of the entire plant
5. Annual load characteristic curve of the heat load demand of the entire plant
6. Structure of the heating network
7. Definition of the pipe system
8. Heat production costs.

Other important tasks, which usually take place at the same time and are not directly related to the technical clarification, are:

- Marketing and customer care
- Heat customer acquisition and connection agreements
- Contract law
- Cost management (business plan and profit and loss account).

The points and work listed are not always firmly assigned to a phase. In some cases, the work must be continued, updated, and concretized across phases. Depending on the project, the process may need to be adjusted.

The next chapters will discuss the individual tasks

6.4.1 Potential heat supply area

A potential heat supply area is, for example, a village, parts of the village, districts, several large customers, or a single large customer with surrounding quarters. The potential heat supply area is divided into zones and areas on the basis of the expected heat reference density of building types (single-family house settlement, industry, apartement buildings etc.) or of geographical conditions such as roads, railway lines or bodies of water. The areas can be simplified and treated like large customers. The local map or, if available, an energy register also serves as a tool for the classification of a location.

It is important to identify the key customers in an area and to attract them to the district heating network. Key customers are characterized by a relatively high power and energy demand (e.g. > 50 kW at 2000 h/a). Examples:

- Industrial areas with process heat
- Areas with multi-family housing or multi-dwelling units (MDU)
- Village or town centers
- Densely built-up areas.

A zone or area can only be optimally and economically connected to a district heating network with key customers, which is why the following procedure is recommended:

1. Define key customers in a potential area
2. Bind key customers contractually (declaration of intent provides planning security)
3. Define the potential routing (network structure)
4. Acquisition of additional potential customers.

The **heat demand density** is a measure of the suitability of a zone for connection to a district heating network (see also Table 6.3), where the annual heat consumption of all buildings is in relation to the property area of the zone:

$$\text{Heat demand density}\left[\frac{\text{kWh}}{\text{a m}^2}\right] = \frac{\text{Annual heat consumption}\left[\frac{\text{kWh}}{\text{a}}\right]}{\text{Property area of the zone}\left[\text{m}^2\right]}$$

The following statements apply for the heat demand density:

- Single-family house settlements are normally not interesting (heat demand density 15 – 30 kWh/a m^2).
- Interesting districts are MDU areas, village or town centers or densely built-up areas.
- Cost efficiency of a district heating network can be maximised if key customers in the zones or in the vicinity can be integrated.
- A district heating network for only one large customer is only interesting if the surrounding areas have a high heat demand density.
- Large customers should be connected via a district heating network if they are located close together.

Table 6.3 Recommended heat demand density as input criteria [21].

Suitability for district heating connection	Heat Demand Density kWh/(a m²)
Not eligible	< 50
Partly eligible	50 – 70
Eligible	> 70

Depending on the situation, it is also possible to efficiently supply areas with a heat demand density below 70 kWh/(a m²) at low cost of investment and fuel, as shown in Table 6.3. However, this must be justified by a budgeted balance sheet and a budgeted income statement (full cost accounting).

6.4.2 Key customer survey

Key customer surveys serve to identify connection interest, and to provide planning data. Suitable data is often not available for existing buildings or special buildings, or gathering it is cumbersome. The effort should be kept to a minimum in the preliminary study, if possible by means of personal enquiries or estimations.

A heat supply area's annual heating demand density for space heating and hot water of individual buildings can also be roughly estimated. The following methods are used for this purpose:

a) Estimation of the annual heating demand, taking into account the energy reference area (ERA) and the building quality. For this purpose, the specific heating demand and the specific heat demand for domestic hot water is added and multiplied by the energy reference area as presented in Table 6.4:

$$Q_{Bldg} = ERA \left(q_h + q_{HW}\right)$$

Q_{Bldg} Annual heat demand buildings in kWh/a

ERA energy reference area in m^2

q_H specific heat demand kWh/(a m^2)

q_{HW} specific heat demand domestic hot water kWh/(a m^2)

b) Estimation of the annual heat demand based on building volume and building quality. For this purpose, the specific heat load demand of a building is multiplied as shown in Table 6.5 by the heated footprint of a building, the floor-to-floor height (visual inspection on site) and the full-load operating hours, see Table 6.6. The number of full-load operating hours of school facilities, industry, commerce and offices with weekend reduction and holiday reduction is reduced accordingly by approximately 15 %.

The load characteristic curve and the annual load-duration curve for the potential heat supply area are determined based on surveys and additional heat demand estimates. The freely accessible Excel table for situation assessment of “QM wood-fired plants” is recommended as a helpful tool [21].

Table 6.4 Specific heating demand and specific heat demand for domestic hot water for buildings of different heating limits and regions of Switzerland [23].

Location	Specific heating demand kWh/(a m^2)			DHW kWh/(a m^2)
Heating limit	15 °C	13 °C	11 °C	–
Middle land (Zurich) 300…800 meter above sea level	60–100	30–60	20–40	20
Mountain Area (Davos) 800…1'800 meter above sea level	75–120	35–75	25–45	20
South Switzerland (Locarno) 200…600 meter above sea level	50–85	25–50	20–35	20

Table 6.5 Specific heat load demand of different building types (based on daily average value without taking into account heating peaks) [21].

Building type	Specific heat load demand W/m^3
Conventionally thermally insulated residential buildings	20 – 27
Well-insulated existing residential buildings	15 – 20
New buildings according to current regulations	8 – 15
Conventional service buildings	23 – 30
Workshop, production rooms or warehouses	10 – 20

Table 6.6 Full-load operating hours for existing buildings (space heating and domestic hot water). These are based on many years of experience for buildings with different heating limits and regions within Switzerland [23].

Space	Full-load operating hours [h/a]		
Heating Limit	15 °C	13 °C	11 °C
Middle land (Zurich) 300…800 meter above sea level	2'000 – 2'500	1'600 – 2'000	1'300 – 1'700
Mountain Area (Davos) 800…1'800 meter above sea level	2'300 – 2'800	1'900 – 2'300	1'600 – 2'000
South Switzerland (Locarno) 200…600 meter above sea level	1'700 – 2'200	1'400 – 1'700	1'400 – 1'800

6.4.3 Updating the heat supply area

In order not to jeopardize the economic operation of the heat distribution network, unsuitable areas with a heat demand density < 50 kWh/a m^2 are excluded in a first step. What remains is the possible heat supply area.

Connected zones with high heat demand density are selected for development. The location of the heat generation and heat network is designed in such a way that the pipeline network remains as short as possible. If some customers cannot be connected because they are unfavourably located in the heat network (route), despite connection interest, these may have to be eliminated or only taken into account for a later extension phase. The original potential heat supply area leaves the heat supply area suitable for development.

The annual heat demand is estimated by assuming a **service connection percentage** of between 50% and 80% of the respective heat supply area.

A **concurrency factor** must also be taken into account for the heat load demand. This describes the effect that a large number of customers combined almost never receive the maximum power at the same time. This effect, known as concurrency, is a basic value for the dimensioning of the heat distribution network and the design of heat generators. The concurrency factor describes the ratio between the maximum simultaneous heat demand of all customers and the total subscribed connection load [40]:

$$g = \frac{\sum_{i=1}^{n} \dot{Q}_i(t_{max})}{\sum_{i=1}^{n} \dot{Q}_{N\_i}(t)}$$

g	Concurrency factor
$\dot{Q}_i(t_{max})$	Heat consumer output i at the time of the maximum output requirement in kW
$\dot{Q}_{N\_i}(t)$	Subscribed rated output of the heat consumer i in kW
n	Number of heat consumers

The concurrency factor is therefore g ≤ 1. In Figure 6.2 the concurrency factor is presented as an approximation function depending on the number of customers, based on a 2001 survey [40]. According to this, 10 to 20 customers can be expected to be in concurrency of about 95% in the scattering range between 85% and 100%. When connected to more than 100 customers, concurrency can be assumed to be approximately 60%.

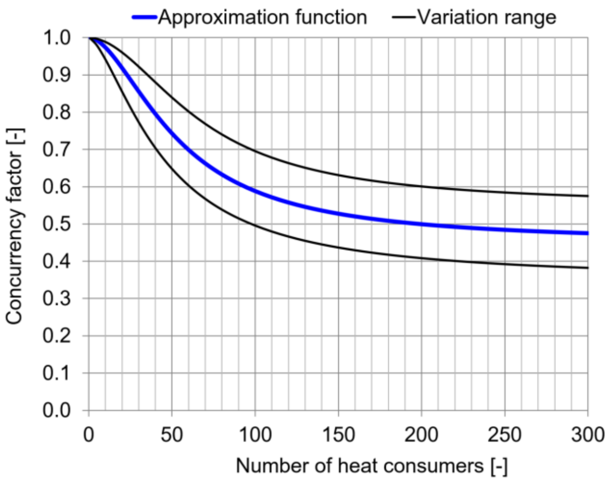


Figure 6.2 Proximity function and scattering range for concurrency factor approximation based on the number of customers [40].

The **customer structure** is decisive for determining concurrency. For example, permanent process heat consumers in a DHS result in a higher concurrency factor than, for example, a sttlement with single-family homes. Any limitations on the supply of power in the Technical Connection Requirements (TCR) also lead to a higher concurrency factor, while temporary or seasonal consumers reduce simultaneity. On the other hand, a seasonal peak load can lead to increased concurrency factors, for example during the holiday season in winter sports resorts with maximum occupancy of hotels and holiday apartments. The assessment of concurrency is therefore based on a great deal of experience and should not be set too low.

When using the excel table situation assessment of “QM Holzheizwerke” mentioned in chapter 6.4.2, care must be taken to ensure that no concurrency factor is used, since the system selection and the design of the heat generator therein are based on a daily average power requirement which takes into account the power requirement similar to the concurrency factor.

The dimensioning of the individual pipe sections is based on a maximum pressure drop for the individual branches that lies between 250 and 300 Pa/m as described in chapter 7.3.

6.4.4 Initial economic analysis

In the case of district heating networks, efficient operation is assumed if the proceeds from the sale of heat exceed the heat production costs from capital and operating costs. An important indicator for estimating whether a district heating network can be cost-effective is the **linear heat density**. The linear heat density is the ratio between the annual amount of purchased heat in MWh/a, and the total length of the route that consists of main, branch and house connection pipelines in meters:

$$\text{Line heat density}\left[\frac{MWh}{a\,m}\right] = \frac{\text{Annual heat consumption}\left[\frac{MWh}{a}\right]}{\text{Total length of the pipeline route}\,[m]}$$

Since the linear heat density serves to characterize the district heating network, customers that consume directly from the heat station, e.g. for their own usage at the location of the heat station, should not be included.

For a rough assessment without exact knowledge of the boundary conditions, heat supply areas with a linear heat density of < 2 MWh/(a m) in the final capacity stage are generally considered unfavourable.

Table 6.7 Recommended linear heat densities of the heat distribution to meet the target value of the specific investment costs [21].

Extension Level	Linear heat density	
	Favorable conditions MWh/(a m)	Unfavorable conditions MWh/(a m)
Initial extension stage	> 0.7	> 1.4
Final capacity stage	> 1.2	> 2.0

Further framework **conditions** such as achievable revenues and investment aid can also influence profitability. In the case of favorable heat sources or construction conditions, even lower linear heat densities can make profitable operation possible (Table 6.7). However, this should be proven by a planned balance sheet and a budgeted plan income statement (full cost accounting).

Connecting small consumers near or along a route is usually no issue in terms of profitability. However, if a small consumer is far away from the nearest main or branch pipe, this reduces the linear heat density, making a connection unattractive; this would need to be assessed case by case. This customer can, if necessary, be connected at a higher cost and higher heat prices.

Recommendations for linear heat density relating to the **investment costs** can be found in the “QM Holzheizwerke” [21] (Figure 6.3).

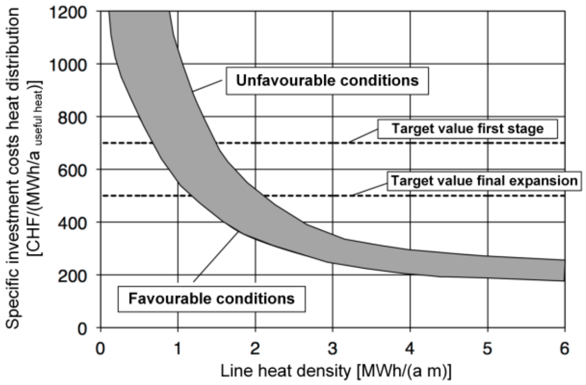


Figure 6.3 Specific investment costs of heat distribution in function of the linear heat density. Range of typical values for [21] (converted in CHF). Included are the costs for the district heating pump groups at the heat station, the costs for the district heating system up to and including the transfer station. Not including costs of consumer installation.

As a recommendation, specific heat distribution investment costs of about 700 CHF/(MWh/a) should be targeted for the first extension stage. In the final capacity stage these should be about another 500 CHF/(MWh/a). In general, these recommendations should not exceed 25%.

In rural areas, the conditions or costs of laying district heating pipes are generally cheaper than in urban areas. In addition, however, the nature of the subsoil may also create additional costs (Table 6.8).

Table 6.8 Impact of framework conditions on the specific investment costs of heat distribution and economy [21].

	Favorable conditions	Unfavorable conditions
Building obstacles	minor	high
Local price level	low	high
Development progress	rapid	slow
Investment grants	high	low
Recoverable income	high	low

Heat distribution losses are also an important indicator influencing efficiency, and depend on the following factors:

- Dimensioning of pipelines
- Thermal insulation class of the pipelines
- Temperature level of supply and return flow
- Linear heat density
- Operating time (year-round or seasonal operation).

In operation, the heat distribution losses are determined as the difference between the amount of heat supplied by the heat generation to the heat network and the amount of heat obtained by all customers (see also chapter 7.1.4).

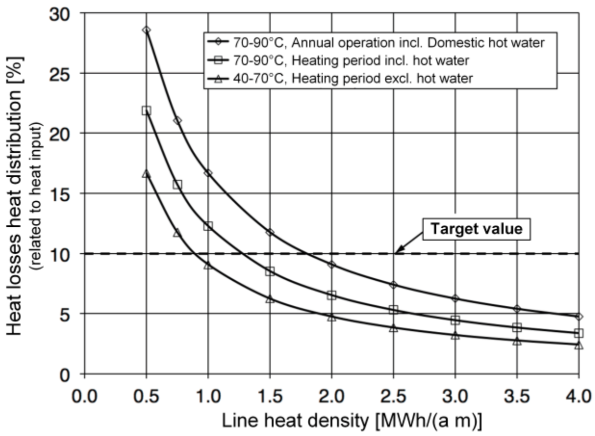


Figure 6.4 Heat distribution losses as function of the linear heat density for different operating modes and supply temperature levels of the heat network according to [21].

Figure 6.4 shows the heat distribution losses as function of the linear heat density for different operating modes and operating medium temperatures. The losses should not exceed a target value of 10% of the usable heat required by the customers. This usually corresponds to a heat distribution loss of about 9% in relation to the heat fed into the grid. In order to meet the target value according to Figure 6.4 the listed linear heat densities have to be complied with, depending on the mode of operation, temperature level, and investment costs as shown in Table 6.9.

Table 6.9 Recommended linear heat density of the heat distribution to meet the target value of the heat distribution losses of ≤ 10 % according to Figure 6.4 [21].

Supply Temperature	Operating Method	Linear Heat Density MWh/(a m)
70-90 °C	Annual operation including domestic hot water	> 1.8
70-90 °C	Heating period including domestic hot water	> 1.3
70-90 °C	Heating period excluding domestic hot water	> 0.8

If the annual heat distribution losses in the final expansion (after climate adjustment) exceed 10%, the first priority should be to try to increase the linear heat density. This not only reduces the heat losses of the network, but also improves overall economic efficiency. If the target value of the heat distribution losses cannot be reached, higher heat losses may be agreed with the developer. It is necessary to examine the long-term impact of the higher heat losses on efficiency, especially with regard to possible increases in fuel prices.

For a first economic assessment of heat production costs, information on the energy offer must be gathered. The offer should be drawn up from the following sources in terms of potential, availability, quality, energy content and price: waste heat, wood, ambient heat, natural gas, heating oil.

Eligible fuel suppliers should provide information on the annual available potential, energy content and fuel price. After calculating the annual fuel demand, the fuel costs can be estimated on the basis of the annual efficiency of heat generation and heat distribution losses.

On the basis of the average **heat production costs** for heat generation, heat distribution and fuel, the individual variants can be compared and assessed in the preliminary study, see chapter 9.3.

An analysis of the annual heat demand and the heat load demand, together with an assessment of the financial situation, is required for a precise assessment of the cost-efficiency of a district heating company. From this situation analysis, heat generation costs and annual costs for an operating period of, for example, 20 years can be reported by using a standardized business plan with a budgeted balance sheet and budgeted income statement.

As a rule, the cost accuracy in phase 1 (preliminary study) is ± 25%

Table 6.10 Checklist Phase 1 – Preliminary study.

Check	Checklist Phase 1 – Preliminary study	Comments
	Potential Heat Supply Area	
	Determine potential key customers > 50 kW (e.g. industry with process heat, public buildings and superstructures in compressed design, etc.)	
	Use of cadastral databases such as webGIS, cantonal energy master plan, etc.	
	Recording heat supply zones with sufficiently high heat demand density (≥ 70 kWh/m²)	
	Capture local and regional energy supply	
	Consider cantonal and communal energy concepts (e.g. cantonal energy structure plans, model regulations of the cantons in the Energy Area MuKEn)	
	Survey of key customers (Is the following applicable?)	
	Connection interest: Yes / No	
	Possible connection time	
	Annual heat demand in kWh/a divided into room heating, hot water and process heat	
	Connection load in kW	
	Energy reference area ERA in m²	
	Required temperature level in °C	
	Update the heat supply area	
	Identify key customers and heat supply area	
	Estimate potential of additional small heat consumers (< 50 kW connection load)	
	Determine load characteristic curve and annual duration curve (e.g. Excel table situation assessment based on "QM Holzheizwerke")	Find additional information under chapter 2.3.
	Evaluate the location of the heat station (possibly several variants)	
	Evaluate the routing of main and branch lines (possibly several variants)	
	Dimensioning of the district heating pipes (max. pressure drop in the section 250 to 300 Pa/m)	Find additional information under chapter 7.3
	Initial economic evaluation	
	Determine the linear heat density	
	Estimate heat distribution losses	Find additional information under chapter 7.1.
	Estimate investment and operating costs (± 25%)	Find additional information under chapter 9.2.
	Estimate heat production costs on the basis of energy supply	See also chapter 9.3.
	Determine necessary tariffs	See also chapter 5.3.3.
	Decision	
	Decision on the further development of the project (iterative process)	
	Transition to conceptual design	

6.5 Phase 2: Conceptual design

In the conceptual design phase, the estimates made in the preliminary study become more specific. The aim is to have one or more economically viable variants to choose from.

For the situation assessment during the conceptual design, a database of the potential customers is created for the calculation and optimisation of the heat network. It is updated and the accuracy increased in the subsequent project steps.

Based on this, the heat supply area is defined (locations for heat stations and routing). A suitable pipe system is evaluated and dimensioned based on the framework conditions.

This is followed by a second profitability review. For further planning security, the highest possible coverage of the annual energy sales should be ensured at the start of construction by means of written assurances (signed heat supply contracts or declarations of intent).

6.5.1 Defining key customers

To survey key customers and other potential heat users, the questionnaire listed in the Appendix (chapter 14) can be used as a template or as is. The questionnaire includes, among other things, the following points:

- Connection interest: Immediately, within the next 5 years, later, no connection.
- Information on the heat load demand and the annual heat demand for space heating, hot water and process heat.
- Information on the current means of heating (fuel, year of construction, efficiency, consumption)
- Information on the hot water preparation system (decentralized electrical, central via boiler, winter and summer operation)
- Information on the domestic heat distribution system (number of heating circuits, radiator, floor heating, temperatures)
- Required temperature level
- Information on renovation plans
- Energy reference area (ERA).

For planning, it is best to create a spreadsheet and record a data set for each customer in the possible heat supply area and check the data for plausibility. The table is a central planning tool, continuously updated and supplemented in the further planning process.

The developer is responsible for allocating a customer status and a connection date. As the project progresses, the highest possible funding ratio for annual energy sales should be ensured by declarations of intent or signed heat supply contracts. As a recommendation, 70% of the annual energy sales should be contractually secured by the start of construction.

6.5.2 Survey minor customers

Potential small heat customers are characterized by the fact that their connection does not negatively affect the linear heat density. The questionnaire can also be used to survey minor heat customers. However, these should only be requested if the key customer is intending so.

6.5.3 Defining the supply area

The heat supply area can be determined based on the key and minor customer survey.

It is now necessary to find a suitable location for the heat station in the supply area, and to design the routing. A location in the center of the supply area is usually economically advantageous to avoid large pipeline cross-sections.

Routing, must avoid changes in direction and junctions. Further information is provided in Chapter 4. The dimensioning of the individual pipe sections is based on a maximum pressure drop in the respective pipe section. This can range between 250 up to 300 Pa/m for the final capacity stage and for the design temperature (see chapter 7.3). Routing must also take into account and clarify the following aspects:

- Transit rights
- Obstacles, crossings (tracks, rivers, etc.)
- Coordination with works management.

6.5.4 Secondary economic analysis

While the economic assessment of the variants in the conceptual design is still based on medium heat generation costs, this is not sufficient for further action, since, for example, losses during the first years of operation are not compensated by profits in later years, but can lead to liquidity problems. For the enterprise to develop, therefore, a budgeted balance sheet and a budgeted income statement for each year of the observation period is required.

This can be performed either the planner or the developer, but the tasks must be clearly defined in the early stages. However, the project-specific data such as the structure and linear heat density of the heating network, which have a major impact on economic efficiency, must be prepared by the planner and made available for the cost-effectiveness assessment. Cost-effectiveness monitoring should not only be carried out during the planning phase, but also while the heating plant is operating, and checked for possible cost optimisations. The cost accuracy in phase 2 (conceptual design) is usually ±15%.

6.5.5 Acquisition

The acquisition of customers is usually the task of the developer or operator of the plant, but it can also be transferred to the planner. It should be noted, however, that the operator is usually solely responsible for marketing, customer support, advising and acquiring new customers after the plant has been handed over. For this reason, it is particularly recommended that new district heating network operators prepare these tasks promptly. It should be noted that marketing and customer acquisition can be time-consuming until the contract is concluded.

Table 6.11 Checklist Phase 2 – Conceptual design.

Check	Checklist Phase 2 – Conceptual design	Comments
	Define key customers	
	Define the connection conditions of key customers (confirm with questionnaire)	Refers to the questionnaire chapter 14
	Check data for plausibility	
	Clarify location, type and scope of the transfer station (e.g. conversion of existing energy centers)	
	Future development of heat demand (renovation intentions, deadlines, etc.)	You should find information out of the questionnaire; refer also to chapter 2.4
	Survey of minor heat consumers (are the following data available?)	
	Connection interest: Yes / No	
	Possible connection time	
	Annual heat demand in kWh/a divided into room heating, hot water and process heat	
	Connection load in kW	
	Energy reference area ERA in m^2	
	Required temperature level in °C	
	Possibly send questionnaire	Refers to the questionnaire chapter 14
	Future development of heat demand (renovation intentions, deadlines, etc.)	You can retrieve further information out of the questionnaire; refer also to chapter 2.4
	Define heat supply area	
	Updating the heat consumers	
	Determination of the heat supply area	
	Update load characteristic and annual duration curve	
	Determine the location of the heat station	
	Define the routing of the main, branch and house connection pipelines (obtain the rights of passage, record obstacles such as road crossings, railway tracks, groundwater, rivers and coordination with service lines)	See also chapter 4.4 until 4.6
	Evaluate the pipe system for main, branch and house connection pipelines (possibly several variants).	See also chapter 4.3.1.7
	Dimensioning of the district heating pipes (max. pressure drop in the section 250 to 300 Pa/m)	See also chapter 7.3.
	Second economic consideration	
	Create/update business plan with budgeted balance sheet and budgeted income statement	See also chapter 9.5
	Determine the linear heat density	
	Estimate Heat Losses	See also chapter 7.1.
	Update local and regional energy supply	
	Estimate investment and operating costs (± 15%)	See also chapter 9.2.
	Update heat production costs on the basis of energy supply	See also chapter 9.3.
	Adjust necessary tariffs if necessary	See also chapter 5.3.3
	Draft heat supply contract and technical connection requirements (TCR)	See also chapter 5.3.
	Decision	
	Decision on the implementation of the project (iterative process)	
	Transition to the tendering and award phase	

6.6 Phase 3: Planning, tendering and awarding of contracts

The aim of Phase 3 is to plan and prepare the respective "district heating project" to the extent that smooth execution is possible. The tasks in Phase 3 include:

- Design of heat network (pipe system, dimensioning, etc.)
- Specify transfer stations
- Create plans
- Prepare tender documents and obtain quotations
- Award contracts in consultation with the developer
- Assisting with economic feasibility

The prerequisite for successful execution is a clear information and communication strategy defined by clear rules. In the rules applicable for services and fees provided by civil engineers according to SIA 103 [79] the rights and obligations of the parties in the conclusion and execution of contracts for engineering services are clearly regulated.

6.6.1 Design heat distribution network

Planning, design, specification and calculation of a heat distribution network:

- Define the execution standard of the heating network
  - Selection of the pipe system and the insulation condition
  - Clarify laying conditions and procedure.
  - Define essential fittings
  - Plan data monitoring and leakage detection.
- Create plans
  - Situation plans 1:200/1:500
  - Longitudinal section
  - Detailed plans of exposed pipelines
  - Trench profiles
- Dimensioning of line sections
- Design of network pumps, valves and other safety devices
- Create pipe statics
- Clarify and, if necessary, obtain rights of transit.

6.6.2 Specification heat transfer station

The concept of heat transfer or the standard for the transfer stations must be defined and the timing must be coordinated with the overall project. All technically relevant points should be listed in the TCR (connection type, the least temperature difference of the heat exchanger, pressure and temperature resistance, safety devices, materials, thermal insulation, data transmission, etc.). A detailed schedule must be drawn up for implementation and agreed with the developer. In exceptional cases, depending on the situation, the requirements can be deviated from, but only if this does not have a negative influence on the return temperature, for example.

6.6.3 Building permit

The aim of the building permit procedure is to obtain permission to build. The dossier must be compiled for submission. Among other things, plans, forms, calculations, etc. are required for this. The building permit procedure must be clarified in advance with the local municipal and state authorities. Depending on the state, the procedure can take up to 6 months.

6.6.4 Tender layout

The services for the pipe and civil engineering works are primarily tendered. It is possible that the installation of the leakage monitoring system as well as the radiographic testing (X-ray) of the welds is tendered separately. Lists of services should be drawn up for calls for tenders, in which general and specific provisions are required. The calls for tenders contain the elements listed below and more or less detailed planning documents:

- Draw up tender plans for civil engineering and pipeline construction
  - Slope
  - Material
  - Covering
  - Special structures (e.g. shafts)
  - Struts
  - Draining and ventilation
  - Sectioning (Shut-off valves)
  - Compensation measures
  - Site measuring
- Instructions for welding the medium pipes, if available procedure test
- General personnel qualificatiosn and proof of qualifications of personnel and contracted companies
- Certificates of steel and plastic welders and work samples, the same for socket fitters included post-insulating.

As a rule, the transfer stations are not integrated within the call for tenders for the district heating network. Depending on the scope of the contract, a separate tender may be required. In this case, a clear specification must be drawn up.

The catalogue of standard descriptions (CSD) [78] makes it possible to standardize and simplify the call for tender documents. The CSD is divided into about 200 chapters, all of which are structured according to the same system and are used to create clear and detailed performance descriptions during construction. It is also a reference book and a checklist for deviations. Chapter 4.14 has been available in the CSD since the beginning of 2017, especially for cables and fittings in the district heating sector.

6.6.5 Submission

In the case of public-law clients (developer), the procedural method must be clarified in advance, and in the case of private clients, the lists of contractors are usually revised together with client. The submission phase ends with checking and comparing the offers and a vendor recommendation for the customer. A bidding round is sometimes carried out in the case of private tenders.

6.6.6 Third economic evaluation

The budgeted balance sheet and the budgeted income statement are updated. The investment and operating costs as well as the heat production costs can be determined. On this basis, the tariffs, some of which are already in force, should be checked. The cost accuracy in phase 3 is around ±10%.

6.6.7 Approval of tender

Based on the submitted offers, the individual work packages can be distributed and awarded. With the completion of phase 3, the execution phase begins, which ends with the approval of the plant.

Table 6.12 Checklist Phase 3 – Planning, Tender and Award of Contracts

Check	Checklist Phase 3 – Planning, tendering and awarding of contracts	Comments
	Design heat distribution network	
	Define the execution standard of the heat network (pipe system, thermal insulation class, installation methods and procedures, fittings, data transmission, leakage monitoring, etc.)	Find additional information within the following chapters 4.3.1.7 and 4.4 up to 4.6
	Define the dimensioning of the district heating pipes (max. pressure drop in the section 250 to 300 Pa/m)	See also chapter 7.3.
	Design of network pumps, fittings and safety devices	Find additional information within the following chapters 7.4, 4.3.3, 8.1.5 and 8.1.6.
	Create pipe statics (calculation of supports, required covering, compensation, anchor points, expansion, verification of static load capacity, etc.)	See also chapter 7.5.
	Specification heat transfer	
	Least temperature difference of the heat exchanger	See also chapter 8.1.9.
	Hydraulic system integration on the primary side between supplier and customer and on the secondary side for room heating, hot water and process heat	See also chapter 8.4 up to 8.6
	Define control and regulation requirements	See also chapter 8.1.7.
	Heat meters (data transmission and monitoring)	See also chapter 8.1.4.
	Building permit	
	Preparation of the documents for the building application	
	Clarify building application with local municipal and cantonal authorities	
	obtaining the building permit	
	Tender layout	
	Drawing up tender plans for civil engineering and pipeline construction (gradients, materials, overburden heights, special structures, bracing, draining, venting, sectioning and compensation measures)	
	Define General Personnel and Company Qualifications	
	Possibly separate tender for transfer station	
	Submission	
	Clarifying the type of procedure (public, by invitation, etc.)	
	Civil engineering and pipeline construction	
	Heat transfer station and hydraulic system integration	
	Measurement and control technology, I&S, control system, etc.	
	Heat station (heat generation, distribution, fuel storage and disposition, etc.)	
	Third economic evaluation	
	Update business plan with budgeted balance sheet and budgeted income statement	See also chapter 9.5.
	Determine investment and operating costs (± 10%)	See also chapter 9.2.
	Determine heat production costs	See also chapter 9.3.
	Control tariffs	See also chapter 5.3.3.
	Award	
	Comparison of offers and awarding of contracts	

6.7 Phase 4: Execution and Approval

In phase 4, the building of the district heating network begins. On the basis of detailed execution plans, trenches are excavated; pipes are laid, welded and inspected. Finally, first subsystems and lastly the entire system is tested for functionality. Defects and malfunctions are recorded in detail and corrected before the plant is put into operation. In this phase, too, a clear information and communication strategy is a prerequisite for successful fulfilment of the order and should be defined by clear rules. As already mentioned in phase 3, the rules for services and fees of civil engineers according to SIA 103 [79] clearly regulate the rights and obligations of the parties in the conclusion and execution of contracts for engineering services.

The completion is the approval or handover of the system to the developer.

6.7.1 Executive plans

On the basis of the tender plans, detailed implementation plans are drawn up or adjusted according to the status and level of detail.

6.7.2 Execution

Construction supervision is essential during execution. This ensures timely and professional execution as well as the overall project coordination. These include:

- Organization of the construction site
  - Organize and record initial meeting or inspections with authorities and contractors on site
  - Arrange and coordinate appointments
- Coordinate and lead construction meetings (project status, interfaces of individual suppliers, schedule, etc.)
- Inform the developer about the progress of the project
- Check the designs, possibly approval of individual subsystems (find checklists in the "AGFW Arbeitsblatt FW 401" Part 17 [100]).
- Earth-laid pipelines should be secured by a route warning tape above each pipeline. Nonmetallic pipes should be installed with a detectable route warning tape.

6.7.3 Commissioning

The following procedure is recommended for commissioning:

- Function control of plant components and systems
- Cold commissioning of the plant
- Eliminate defects and malfunctions
- Commissioning of the plant
- To draw up a protocol on the course and result of the commissioning and to sign it by all parties involved.
- Any defects that occur must be remedied by the time of handover.

With commissioning, the responsibility is generally transferred to the operator. However, a district heating network is often put into operation before all the construction work (e.g. restoration of the surface) has been completed, in which case the responsibility of the construction site lies until the work is completed or until the official handover to the operator.

6.7.4 Documentation

The prerequisite for safe operation and maintenance is reliable, complete, up-to-date pipeline documentation. This contributes significantly to the fact that the pipelines and plant parts can be found quickly and reliably, and forms the basis for planning expansion and maintenance projects according to [6]. It also helps to render detailed plant documentation. The basis for reliable pipeline documentation is precise measurements and document tracking. The cost of tracking can be considerable and must be planned for accordingly.

The scope of documentation for the technical part can be divided into two main areas:

- The test and verification documentation
- The operational technical documentation.

The **test and proof documentation** consists of up-to-date documents about the plants and provides information about their proper installation. They include the following:

- Certificates of manufacture (planning documents, quality inspections, approval certificates)
- Construction site documentation as proof of quality control and warranty claims
- Extensions (new building, customer connections)
- Amendments to documentation to include changes to technical systems during the course of maintenance (replacement of components or system parts)
- Changes in technical installations due to official or private requirements (route reallocation, contractual changes).

An overview of the necessary documentation concerning quality assurance measures includes the following points:

- Approvals and associated records of deliveries and services
- CE declarations including hazard analysis
- Levelling of the district heating pipeline
- Billing isometrics for pipe construction
- Delivery and verification of the pipe manufacturer
- Overview of the material certificates with item allocations to steel pipe parts, casing pipes, sleeves, PUR foam, fittings, etc.
- Evidence and results of internal and external plant inspections
- Films and protocols for radiographing the factory welding seams
- Records of:
  - Installation work and quality control

  - Welding work on the carrier pipe
  - Prestress or preheating of the district heating pipe
  - Post-insulation work and installation of expansion pads
  - Loop measurement with the leakage monitoring system
- Calculation verification (revision status)
- Plans, drawings and static proof of shaft structures
- Static proof of pressure- and protection pipes, so required
- Entire route plan overview
- Revised route and detail plans
- Revised plan of the leakage monitoring system in connection with the accompanying cable routing (geographical representation)
- Parts drawings for fittings, mountings, etc. for spares inventory.

The **operational documentation** includes all documents that contribute to safeguarding and supporting the proper management of the company as well as to the observance of occupational safety:

- Regulations, laws, standards
- Operating regulations and work instructions
- Operating manuals
- Documentation of periodic inspections
- Organizational plans (responsibility structures, preparedness plans, emergency and deployment plans, training plans)
- Service and maintenance planning
- Life cycle documentation for plants (damage analyses, fault and repair reports, etc.)
- Device lists with monitoring and test certificates (electrical devices, heat meters, manometers, temperature sensors, etc.)
- Customer files
- Equipment and equipment labelling systems.

All necessary documents in the company must be kept accessible and taught in special training courses.

In addition to the documentation in written form (plan documents, etc.), the marking of the plant parts on site must also be taken into account. The labels attached to the objects should therefore correspond to the designation in the plan. In addition, district heating transmission systems must also be designated in public spaces. The following should be noted:

- Above-ground structures and operating points in overhead lines should be equipped with the system labels and plain text.
- For underground structures, shaft covers should be used which indicate a district heating system.
- Built-in underground valves in underground pipelines must be designated using signs similar to gas or water.

6.7.5 Cost control

Execution and approval also include a construction invoice, in which all costs for the construction of the district heating network are collected and recorded in a report.

6.7.6 Approval

Once the plant has been approved, the entire system is formally handed over to the developer. At the very least the responsible planner, the plant supplier and the developer must be present at site approval. Approval is based on:

- Approving the requirements specifications
- Check documentation and update if necessary
- Approving cost control
- Create an approval report with signatures of all parties.

Table 6.13 Checklist Phase 4 – Execution and approval

Check	Checklist Phase 4 – Execution and approval	Comments
	Execution plans	
	Drawing up implementation plans for civil engineering and pipeline construction on the basis of the tender plans (gradients, materials, pipe depth, special structures, bracing, draining, ventilation, sectioning and compensation measures)	
	Execution	
	Conclusion of a civil engineering insurance policy	See also chapter 6.9.4.
	Construction supervision during the construction phase (control of the on-schedule and professional execution)	
	Coordinate and lead construction meetings (project status, interfaces of individual suppliers, schedule, etc.)	
	Informing the developer about the project status	
	Inspection of the designs, possible approval of individual subsystems	
	Commissioning	
	Function control of all plant components and systems	
	Warm start-up of the plant	
	Eliminate defects and malfunctions	
	Commissioning of the plant	
	Documentation	
	Create test and verification documentation	
	Preparation of operational documentation	
	Create an operating manual	
	Cost control	
	Create a construction accounting in which all costs for the construction of the district heating network are collected.	
	Approval	
	Approval of the entire system on the basis of the specifications. The planner, the contractor(s) and the developer should be present at the approval.	
	Review and update documentation	
	Update operating manual	
	Instructing the plant operator	
	Create an approval report and have it signed by all parties.	

6.8 Phase 5: Operational optimisation

The traditional methods of commissioning and approval, especially for complex plants, are often inadequate and are not a guarantee for the economic operation of the plant. Even if plants have been properly designed and built, they are often not operated in the way envisaged in the concept. Operational optimisation is indispensable for proper functioning, so that plant functions are systematically checked and compared with the specifications in the function description after the plant has been handed over to the developer. This check can address deficiencies in plant components and improve the control parameter settings.

In the case of subsidized projects, operational optimisation can be an essential part of the subsidy conditions and can therefore also be compulsory.

Depending on the size of the plant, operational optimisation is carried out in the first one to two years after plant approval. Depending on the degree of expansion and the planned final capacity, operational optimisation can or should be repeated or caught up on. In the case of district heating systems, operational optimisation should be carried out after at least one full heating period.

For targeted data collection, it is necessary to develop a concept for operational optimisation early in the planning process. The concept regulates when, by whom and what operating data is collected and analyzed. The operational optimisation concept, in particular its sequence of implementation, must be signed by the developer and the responsible planner. Data must be collected from commissioning of the plant onwards and continued as long as contractually agreed. However, data collected for operational optimisation should be for at least one year.

In order for a successful operational optimisation to be carried out together with the plant suppliers, appropriate financial guarantees for the warranty period must be provided as stated in the service contracts (liability rebates), otherwise not all plant suppliers would participate in the operational optimisation.

6.8.1 Data collection

As mentioned above, operational optimisation examines whether the plant is working as intended during the design. This can only be carried out if it has been specified in advance how the installation should work. The necessary prerequisites for operational optimisation are on the one hand a concept for operational optimisation and on the other hand the following conditions:

- Function description with pictorial schematic which describes in detail how to run the system under various operating conditions.
- List of the agreed and guaranteed key figures that prove optimal operation.
- Measuring site list in which the measuring points, the measuring range, the resolution and the measuring accuracy are specified for each measuring point.
- Description of how automatic data recording is performed. The responsibilities must also be recorded here:
    - Specification and planning of data monitoring
    - Data retrieval
    - Data Analysis
- The plant must be equipped with the appropriate control technology and be able to record and archive the required data.

Remote access for the planner and the plant supplier is of great advantage for quick access to the operating data. With continuous data collecting, e.g. with a control system, data can also be analyzed and optimisations made at a later time.

By means of data communication in the district heating network and data collection in a central control system, the data of the respective in-house transfer stations is constantly accessible to the heat control center. This logs all relevant data of the transfer stations (momentary power, flow rate, temperatures, target temperatures, operating status, pump and valve status, etc.); it can be displayed and analyzed graphically.

This data communication is established by a data bus. The data transfer range must not be limited by the number of controlled devices connected. The hardware design of the interfaces must be designed for the conditions within the district heating environment (possible interference due to parallel installed energy cables, different soil potentials, danger from direct and indirect lightning strike).

The continuous data network makes the processes in the entire district heating network transparent and it is possible to remotely adjust all plant parameters of each individual transfer station. Customers can therefore be supported from any location (e.g. also by notebook and mobile phone) and the system adjusted accordingly.

Cables are usually laid in a tree shape with a loop-in to each building. Therefore, no loops, rings or terminators in the network are necessary, whereby cable sockets in the ground are to be avoided. The data cables to be laid should be multi-pole (according to the requirements of the system supplier) and shielded. Before the individual district heating controllers are connected to the data cable, it is essential to check the entire cabling for interruptions or short circuits. In order to have detailed documentation of the cable laying available for later network extensions and for possible troubleshooting, the location of the cables must be documented precisely.

6.8.2 Analysis

A graphical representation of the data is essential for different operating states so that the data collected can be interpreted and analyzed. The minimal operating conditions to be examined are the low load operation (spring, autumn or summer time) and the operating conditions in cold weather. The data of the two operating states should meet the following requirements:

- Representation of weekly progressions

- Representation of daily progressions (24h progression) on selected days
- Display all of the most important data in a single chart
- Uniform division and labelling of the axes
- No more than six parameters in a chart

Analysis of the recorded operating data is based on the following criteria:

- Set point/actual comparison with reference values (temperature and pressure level according to TCR)
- Interpreting deviations
- Determining optimisation potential
- Error and damage documentation

6.8.3 Optimisation

Optimisation measures are based on the analysis of the collected data. From this an action plan is drawn up, which must be approved and implemented by the client. The measures can also be implemented in stages, depending on the priority.

The optimisations made must always be checked and, if necessary, readjusted. The most common adjustments or optimisation measures are:

- Hydraulic system adjustment
- Setting the setpoints and controller parameters (temperature, pressure, etc.)
- Program the time settings

Where possible, recording and analyzing the most important key values of the customers (e.g. energy in kWh and water quantity in m3) should be continued. This ensures that possible faults and optimisation potentials are detected quickly and easily (see also Chapter 10).

Table 6.14 Checklist Phase 5 – Operational optimisation

Check	Checklist Phase 5 - Operational optimisation	Comments
	Data collection	
	Creation of a concept for data collection (usually obligatory for subsidized projects)	
	Recording and recording plant data	
	Log setting of reference values	
	Analysis	
	Graphical processing of the data	
	Target/Actual Comparison with Reference Values	
	Interpreting the Deviations	
	Determination of the optimisation potential	
	Error and damage documentation	
	Optimisation	
	Create action plan for improvement (implementation)	
	implement measures	
	follow-up check	

6.9 Phase 6: Operation and Management

6.9.1 Operational concept

Before the end of the warranty period, if possible during final inspection, the head planner must prepare a company, training and maintenance concept in addition to updating the documentation and hand it over to the client. This concept must include the following elements:

- Revised data sheets for manual acquisition of operating data
- Information on how operating personnel must be trained and educated to carry out their duties.
- revised maintenance plan
- Alerting and on-call service
- Emergency plan
- Performance monitoring.

Performance monitoring ensures that the system meets the requirements for optimal operation even after the warranty period has expired. The operating concept must therefore specify:

- What data is recorded how (manually, automatically)
- How this data is analyzed (for example, which key figures, which Excel evaluations)
- Who is responsible for the evaluation and interpretation of the results?
- Who is to be contacted in the event of malfunction or incidents?
- The intervals at which periodic inspections are to be carried out
- Which operating resources are to be analyzed at which intervals?

6.9.2 Maintenance

Maintenance means taking measures to maintain and restore the target state, as well as to determine and assess the actual state of a system. These measures include:

- Inspection
- Maintenance
- Repairs.

The repair objectives must be aligned with the company's objectives and appropriate maintenance strategies must be defined.

In order to ensure trouble-free operation, the maintenance strategy should not be limited to so-called event-oriented maintenance, i.e. troubleshooting. Preventive maintenance based on operating results, monitoring and inspection is recommended. Regular inspections and targeted maintenance of all important systems and plant components are of prime importance.

6.9.3 Maintenance contract

Depending on the size and complexity of the plant, as well as the number of companies involved, a maintenance contract must be agreed with the plant suppliers, in particular the heat supply. In this way, faults can be reduced to a minimum and operational safety can be maximised. Possible contractual partners are:

- Supplier of boiler with control system
- Supplier of fuel supply
- Supplier of exhaust gas treatment
- Supplier of the hydraulic system (piping)
- Supplier of the control or guidance system
- Supplier of the leakage monitoring system
- Supplier of the transfer stations.

A maintenance contract may have to be concluded with all the above-mentioned suppliers. The included services should be similar or the same for all.

The following points should be included in a maintenance contract:

- Purpose of the contract
- Clear description of scope of delivery and performance (guarantees also possible!)
- Description of exclusions and exceptions
- Hourly rates and incremental costs, expenses
- Costs
- Validity and duration
- Termination
- Extension
- Contact address and procedure for organizing assistance in emergencies
- Operator’s obligations
- Owner’s rights
- Place, date and signature of client and contractor

6.9.4 Insurance

Appropriate insurance coverage must also be considered to secure long-term economic operation of a district heating company. This is particularly important because contractual heat supply obligations are largely towards the customers. No heating plant can be guaranteed to perform free of unexpected system malfunctions or interruptions, or to always supply economical heat.

When taking out insurance, the following products should be considered:

- Construction insurance (during construction phase)
- Fire insurance
- Machine breakage and operational interruption insurance
- Business liability insurance.

As the insurance sector is relatively large and complex, it is advisable to carry out a risk assessment and to consult experts (e.g. insurance brokers).

Table 6.15 Checklist Phase 6 – Operation and management

Check	Checklist Phase 6 – Operation and management	Comments
	Operational concept	
	Create operational concept	
	Recording of the operating data	
	Define training and further training measures for operating personnel	
	Create maintenance plan	
	Alerting and on-call organization	
	contingency plan	
	Conduct performance monitoring	
	Maintenance	
	Define maintenance strategy (preventive maintenance on the basis of operating results and control and inspection work)	
	Maintenance Contract	
	Agree maintenance contracts with suppliers of the most important plant systems.	
	Insurance	
	Carry out risk assessment and call in experts if necessary (e.g. insurance brokers)	
	Fire insurance	
	Machinery breakdown and business interruption insurance	
	Public liability insurance	

7 Heat Distribution – Calculation

7.1 Heat loss

District heating distribution heat loss is influenced by the pipe system (material, dimensioning, thermal insulation class, laying, etc.) and the operating conditions of the district heating network (temperature level, temperature difference, operating regime, etc.).

There are basically two approaches to determining heat loss.

First, heat loss can be determined by means of heat meter data by comparing the amount of heat fed into the grid each year with the total amount of heat supplied to the customers. Radiation and convection losses from transfer stations, pumps, fittings, etc. are also taken into account. With this method, however, the heat loss can only be measured once the plant goes into operation.

Secondly, heat loss can be estimated using network plans and data from the pipe systems used. Compared to the first method, the second method typically has a lower value, since the abovementioned convection and radiation losses are usually not taken into account.

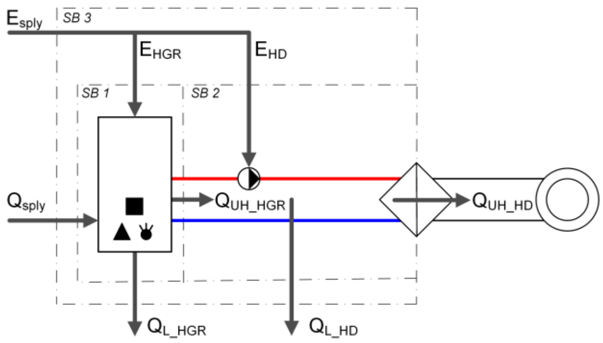


Figure 7.1 System boundaries for the evaluation of a district heating network based on the terms described by Table 7.1.
SB1: Heat generation
SB2: Heat distribution primary side
SB3: District heating system (heat generation and distribution).

The boundaries presented in Figure 7.1 help to assess a district heating network (Table 7.1). The system limit SB 1 represents heat generation (HG). On the one hand, the energy flow is electrical energy required for heat generation units such as pumps, conveying systems, ventilators etc. and on the other hand, the heat contained in the fuel. The energy flow discharged from the system boundary SB 1 consists of heat loss (incomplete combustion and radiation losses of the heat generation plant) and the amount of heat generated for use.

The system boundary SB2 represents the heat distribution (HD). Supplied energy flow represents the electrical energy required for the heat distribution units, such as pumps, control valves, frequency inverters, leakage monitoring, etc. and the heat fed into the district heating network from the heat generation. The energy flow discharged from the SB2 system limit consist of heat loss from the heat distribution network and the amount of heat sold to the customers.

The system boundary SB3 represents the entire district heating system. Supplied energy flow is the electrical energy for heat distribution units, heat generation and the heat contained in the fuel. The energy flow discharged from the SB3 system limit consists of heat loss from heat generation, heat loss from heat distribution and the heat sold to the customers.

Table 7.1 Key to representation of system boundaries Figure 7.1

Description	Symbol	Unit
Supplied electricity quantity	E_{SPLY}	MWh/a
Electric energy quantity heat generation	E_{HG}	MWh/a
Electric energy quantity heat distribution	E_{HD}	MWh/a
Heat supplied	Q_{SPLY}	MWh/a
Usable heat generation	Q_{UH_HG}	MWh/a
Usable heat distribution	Q_{UH_HD}	MWh/a
Heat loss heat generation	Q_{HL_HG}	MWh/a
Heat loss heat distribution	Q_{HL_HD}	MWh/a

Figure 7.2 represents a district heating network based on the target values according to “QM Holzheizwerke“ [21] and Figure 7.3 depicts an overdimensioned district heating network taken from a practical survey [16]. The illustration shows that overdimensioning reduces the electrical from 2 % to 1.35 %, but that the heat losses rise from 10 % to 16.5% (Table 7.2). The practical study [16] indicates that overdimensioned piping is responsible for the higher heat loss.

Table 7.2 Comparison of proportional heat loss, electrical energy for heat production and heat distribution.

Description	Example QMH	Example Real
Heat loss heat distribution[1]	10 %	16.5 %
Electric energy quantity heat generation[1]	1.25 %	1.1 %
Electric energy quantity heat distribution[1]	0.75 %	0.2 %

1) Reference value is the usable heat of heat generation Q_{UH_HG}

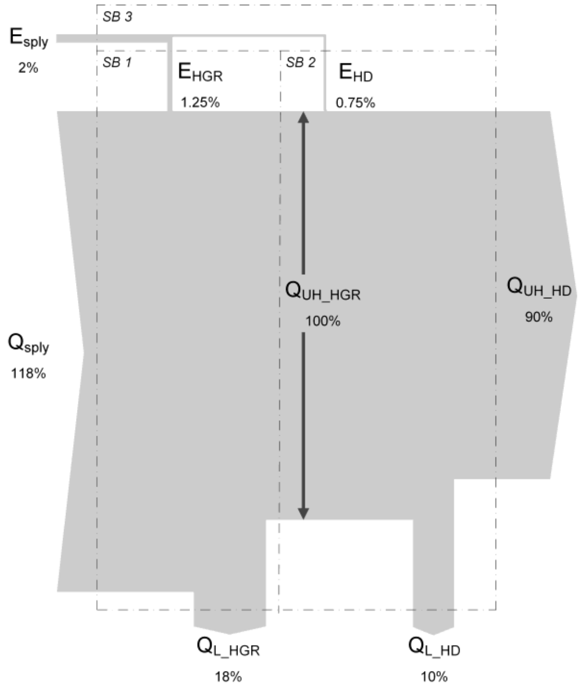


Figure 7.2 Sankey-Diagram for a heat network following an example presented in “QM Holzheizwerke”.

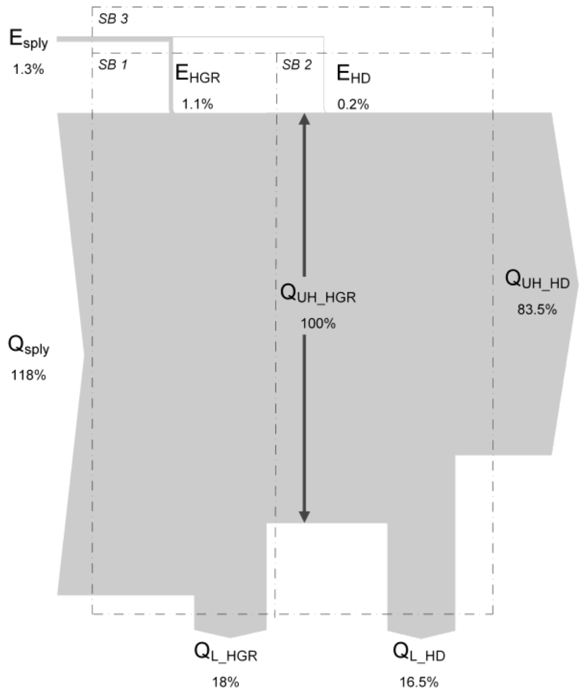


Figure 7.3 Sankey-Diagram for a heat network as a real example.

7.1.1 Overground pipes

For above ground (air-flushed) pipelines, the heat transmission coefficient h for heat-insulated pipes is is based on the inner radius of the medium-transporting inner pipe, as this must be available for hydraulic system calculations anyway.

Figure 7.4 shows the overall heat transfer coefficient U in W/(m² K) calculated with the inner radius as a reference value as follows:

$$U_i = \frac{1}{\frac{1}{\alpha_i} + \frac{r_i}{\lambda_P}\ln\left(\frac{r_P}{r_i}\right) + \frac{r_i}{\lambda_{IM}}\ln\left(\frac{r_{IM}}{r_P}\right) + \frac{r_i}{\lambda_{JP}}\ln\left(\frac{r_{JP}}{r_{IM}}\right) + \frac{r_i}{r_{JP}\ \alpha_a}}$$

With the usual materials, pipe diameters and flow velocities in heating construction, the inner heat transfer coefficient and the thermal conductivity resistance of the pipes can be assumed to be infinitely large for the calculation. For the outer total heat transfer coefficient α_a, consisting of the convection and radiation portions, an average value of approximately 9.7 W/(m² K) can be used for heat-insulated pipes in buildings and ducts. For overhead lines, the heat transfer coefficient depends on the wind speed. The mean value can be approximately 23.2 W/(m² K). The overall heat transfer coefficient U_P is therefore based on the outer radius of the pipe of the medium-transporting pipe r_P. These simplifications result in the following expression [44]:

$$U_P = \frac{1}{\frac{r_P}{\lambda_{IM}}\ln\left(\frac{r_{IM}}{r_P}\right) + \frac{r_P}{r_{IM}\ \alpha_a}}$$

Multiplied by the relevant surface area of the pipeline in m²

$$A_P = 2\pi l r_P$$

and the temperature difference between the inside and outside in K.

$$\Delta T = T_a - T_i$$

The following equation is obtained for the heat loss flow in W:

$$\dot{Q}_L = U_P\ A_P\ \Delta T = \frac{2\pi l (T_a - T_i)}{\frac{1}{\lambda_{IM}}\ln\left(\frac{r_{IM}}{r_P}\right) + \frac{1}{r_{IM}\ \alpha_a}}$$

The specific heat convection losses per meter in W/m result in:

$$\dot{q}_L = \frac{\dot{Q}_L}{l} = \frac{2\pi (T_a - T_i)}{\frac{1}{\lambda_{IM}}\ln\left(\frac{r_{IM}}{r_P}\right) + \frac{1}{r_{IM}\ \alpha_a}}$$

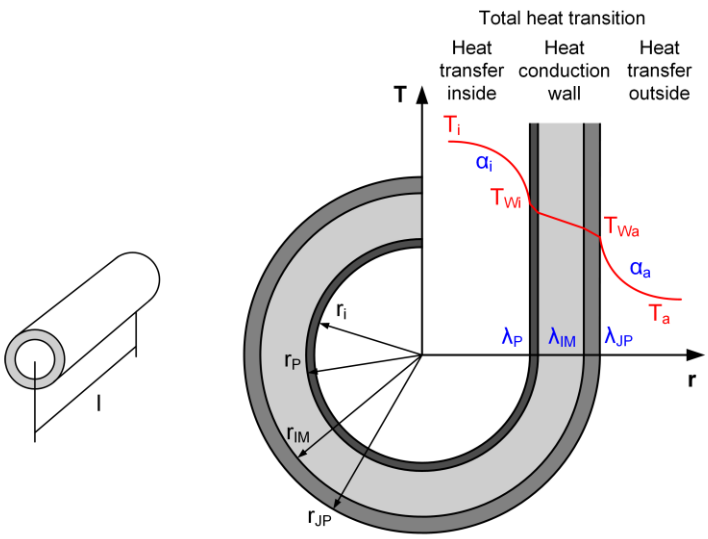


Figure 7.4 General representation of the heat transmission through a thermally insulated pipe.

7.1.2 Earth-laid pipes

Nowadays, district heating networks usually use earth-laid (underground) district heating pipes. Steel or plastic pipes are thermally insulated and encased in a jacket pipe. Calculating heat loss flow is complicated. Thermal conduction works at least two dimensionally and under fringe conditions such as constant temperature at the pipe circumference, constant outside air temperature at the earth's surface and constant groundwater temperatures at a predefined depth. In reality, the pipe temperature and the outside temperature would have to be defined as a time function and hence for non-stationary thermal conduction. The finite element method may be applied for the numerical calculation. However, this is complicated and cannot be generalized and requires knowing the exact temperature regime in the observed network section as a time function [44].

For a good approximation, however, heat transport can be treated as a stationary process because the following applies:

- The insulating material of the pipe reduces approx. 80% to 90% of the temperature difference (pipe to ground). Because the layers are relatively thin, the jacket's temperature profile can be regarded as approximately quasi stationary.
- The network's temporal operation is generally not exactly known. Due to thick damping, short, periodic temperature fluctuations only influence the soil temperature in the immediate vicinity of the pipes, and can be neglected.
- Temporal average values are used to calculate the ambient temperature of the ground surface and the ambient air and groundwater, similar to the standardized calculation used for heat loss in ground-touching components.

For the district heating pipes, which are usually earth-laid today, thermal conduction from the ground to the surrounding area and how the two pipelines influence each other must also be taken into account.

As shown in Figure 7.5, the coverage thickness h_{Cov} and the pipe clearance distance between the two pipelines constitute the minimum thermal barrier of the soil in relation to the surrounding area. As presented in chapter 0, the heat conduction resistance of the pipes (steel and plastic jacket pipes) are neglected.

The total heat transfer coefficient consists of the following three parts:

- Thermal conduction through the insulating material of the pipes
- Thermal conduction through the soil
- How the two pipelines influence each other.

For simplification, from now on earth-laid district heating pipes with pre-insulated steel pipes in single-tube version will be examined. The calculation can also be applied to flexible pipe systems such as pre-insulated flexible steel pipes and pre-insulated PEX pipes, provided that the required dimensions and data are available.

The following restrictions apply to the simplified calculation of the specific heat loss flow for earth-laid district heating pipes with rigid steel pipes in single pipe design:

- Steel pipe
- Plastic jacket pipe (PE)
- Polyurethane foam (PUR) firmly bonded to the medium and jacket pipe as insulation material
- Supply and return pipes have the same nominal diameter
- Can only be calculated individually for one section at a time

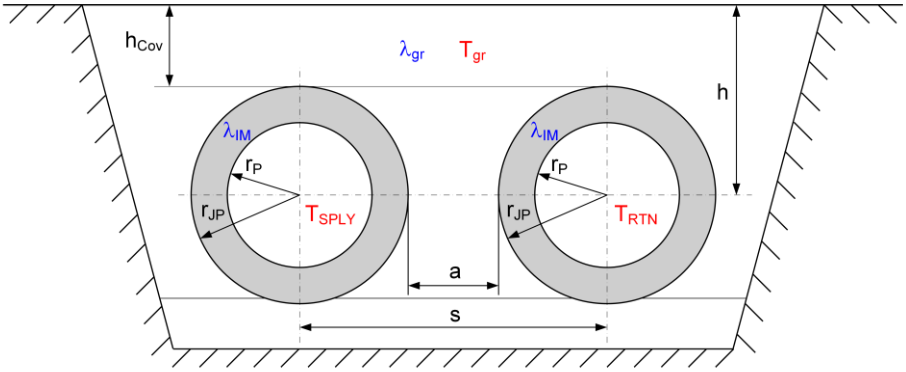


Figure 7.5 Schematic representation of earth-laid district heating pipe in single pipe design.

From [3], the following relationship was assumed for the overall heat transfer coefficient in W/(m² K) of two underground single pipes. However, the overall heat transfer coefficient applies to only one of the two pipes. The reference value is the outer radius of the steel pipe r_R, since the temperature drop in the steel pipe is negligible:

$$U_P = \frac{1}{\underbrace{\frac{r_P}{\lambda_{IM}} \ln\left(\frac{r_{JP}}{r_P}\right)}_{\text{Insulation}} + \underbrace{\frac{r_P}{\lambda_{gr}} \ln\left(\frac{4\,h}{r_{JP}}\right)}_{\text{Ground}} + \underbrace{\frac{r_P}{\lambda_{gr}} \ln\left\{\left[\left(\frac{2\,h}{s}\right)^2 + 1\right]^{0.5}\right\}}_{\text{Mutual Influence}}}$$

The laying depth h in m can be expressed in relation to the minimum coverage thickness h_{Cov}, as follows:

$$h = h_{Cov} + r_{JP}$$

The horizontal pipe distance s in m can be calculated in relation to the pipe clearance distance as follows:

$$s = a + 2\,r_{JP}$$

Inserting the laying depth h and the horizontal pipe distance results in the following heat transfer coefficient in W/(m² K):

$$U_P = \frac{1}{\frac{r_P}{\lambda_{IM}} \ln\left(\frac{r_{JP}}{r_P}\right) + \frac{r_P}{\lambda_{gr}} \ln\left(\frac{4\,(h_{Cov} + r_{JP})}{r_{JP}}\right) + \frac{r_P}{\lambda_{gr}} \ln\left\{\left[\left(\frac{2\,(h_{Cov} + r_{JP})}{a + 2\,r_{JP}}\right)^2 + 1\right]^{0.5}\right\}}$$

The determinative surface area in m² is calculated as in chapter 0, however for both supply and return lines. The reference value is the outer radius of the steel pipe r_R.

$$A_{S/R} = 2\,2\,\pi\,L\,r_P = 4\,\pi\,L\,r_P$$

The temperature difference in earth-laid pipes in K is calculated based on the difference between the designed operating medium temperature T_{OP} and the mean soil temperature T_{gr}:

$$\Delta T_{ELP} = T_{\overline{OP}} - T_{gr} = \frac{T_{SPLY} + T_{RTN}}{2} - T_{gr}$$

The heat loss flow in the supply and return flow of a partial section in W is calculated based on the last three equations as follows:

$$\dot{Q}_L = U_P\,A_{S/R}\,\Delta T_{ELP}$$

$$\dot{Q}_L = \frac{4\,\pi\,L\left(\frac{T_{SPLY} + T_{RTN}}{2} - T_{gr}\right)}{\frac{1}{\lambda_{IM}} \ln\left(\frac{r_{JP}}{r_P}\right) + \frac{1}{\lambda_{gr}} \ln\left(\frac{4\,(h_{Cov} + r_{JP})}{r_{JP}}\right) + \frac{1}{\lambda_{gr}} \ln\left\{\left[\left(\frac{2\,(h_{Cov} + r_{JP})}{a + 2\,r_{JP}}\right)^2 + 1\right]^{0.5}\right\}}$$

The specific heat loss flow per routed meter in W/m results in:

$$\dot{q}_L = \frac{\dot{Q}_L}{L}$$

$$\dot{q}_L = \frac{4\,\pi\left(\frac{T_{SPLY} + T_{RTN}}{2} - T_{gr}\right)}{\frac{1}{\lambda_{IM}}\ln\left(\frac{r_{JP}}{r_P}\right) + \frac{1}{\lambda_{gr}}\ln\left(\frac{4\left(h_{Cov} + r_{JP}\right)}{r_{JP}}\right) + \frac{1}{\lambda_{gr}}\ln\left\{\left[\left(\frac{2\left(h_{Cov} + r_{JP}\right)}{a + 2\,r_{JP}}\right)^2 + 1\right]^{0.5}\right\}}$$

7.1.3 Specific heat loss per routed meter pipeline

The annual heat loss of a heat network can be calculated in a simplified manner using the specific heat loss per routed meter of the pipeline, the mean operating medium temperature, and the annual operating time of the heat network.

The specific heat loss per routed meter pipeline is calculated using the heat transfer coefficient, the area of the surface, and the length of the pipeline. This information can usually be taken from the product information of the pipe system manufacturers and is given in watts per routed meter and Kelvin [W/(m K)]. In Figure 13.6 the specific heat loss per routed meter pipeline up to DN 200 for pre-insulated rigid and flexible steel pipes and pre-insulated PEX pipes are shown and were calculated according to the equation below. The information on the double pipe versions was taken from the product information of the following companies: Brugg Pipe Systems, Isoplus and Logstor.

$$\dot{q}_{L,ELP} = \frac{U_P\ A_{S/R}}{L} = \frac{U_P\ 4\,\pi\,L\,r_P}{L} = U_P\ 4\,\pi\ r_P$$

$$\dot{q}_{L,ELP} = \frac{4\,\pi}{\frac{1}{\lambda_{IM}}\ln\left(\frac{r_{JP}}{r_P}\right) + \frac{1}{\lambda_{gr}}\ln\left(\frac{4\left(h_{Cov} + r_{JP}\right)}{r_{JP}}\right) + \frac{1}{\lambda_{gr}}\ln\left\{\left[\left(\frac{2\left(h_{Cov} + r_{JP}\right)}{a + 2\,r_{JP}}\right)^2 + 1\right]^{0.5}\right\}}$$

7.1.4 Annual heat loss

The heat loss during heat distribution influences a system's efficiency and depends on the linear heat density, operating time (full-year or seasonal operation), the temperature level, dimensioning and thermal insulation class of the pipeline. The annual heat loss usually relates to the amount of heat fed into the network each year and is expressed as a percentage.

Metered heat loss
During operation, heat loss can be determined on the basis of heat meter data that compares the heat quantity fed annually into the network with the total heat quantity supplied to the heat customers. Radiation and convection losses of transfer stations, pumps and fittings are also taken into consideration. Annual heat loss is calculated based on the difference between the heat quantity fed annually into the network and the total heat quantity supplied to the heat customers in relation to the heat quantity fed annually into the network.

$$q_{L,a} = \frac{\text{annual heat losses of the heat distribution } 100\%}{\text{annual heat input to the distribution network}}$$

Calculated heat loss
Heat loss can be estimated based on network plans and data from the piping system used. Compared with the metered method, the calculation should result in a lower value, because the mentioned convection and radiation losses are usually not be taken into account. To calculate the annual heat loss in the heat distribution system, additional information is required such as the specific heat loss per routed pipeline meter, (see Chapter 7.1.3) and the average operating medium temperature:

$$T_{\overline{OP},m} = \frac{T_{SPLY,m} + T_{RTN,m}}{2}$$

The mean operating medium temperature must be determined with gliding supply temperature in relation to the outside temperature. At constant supply and return temperatures, the mean operating temperature corresponds to the operating medium temperature according to design:

$$T_{\overline{OP}} = \frac{T_{SPLY} + T_{RTN}}{2}$$

The temperature difference to the ground must also be taken into account. The average temperature difference in K for earth-laid pipelines is calculated as the difference between the average operating medium temperature (or the operating temperature according to design) and the mean ground temperature T_{gr}:

$$\Delta T_{ELP,m} = T_{\overline{OP},m} - T_{gr} = \frac{T_{SPLY,m} + T_{RTN,m}}{2} - T_{gr}$$

The annual heat loss for one section in kWh/a is calculated as the specific heat loss per routed meter of pipeline, the average temperature difference for earth-laid pipelines, the length of the routed pipeline section and the annual operating time of the network as follows:

$$Q_{L,a,i} = \frac{\dot{q}_{L,ELP} \; \Delta T_{ELP,m} \; L}{1000} \; \tau_N$$

The annual heat loss of the heat distribution in kWh/a is calculated as the sum of the heat loss of the individual sections:

$$Q_{L,a} = \sum Q_{L,a,i}$$

The annual heat loss also depends on the amount of heat supplied annually into the network and is represented as a percentage:

$$q_{L,a} = \frac{Q_{L,a} \; 100\%}{Q_{N,sply,a}}$$

7.1.5 Temperature drop as function of distance

Due to heat loss in the district heating lines, the heat-transporting medium cools down along the transport distance. This applies to both the supply and return lines. For the district heating operator, the temperature drop in the supply line is particularly important, as a contractually agreed supply temperature must be provided. The temperature drop from the heat station to the most remote customer is calculated - simplified - on the basis of the equation for the heat flow rate as follows:

$$\dot{Q} = \dot{m} \; c_{pW} \; \Delta T$$

Thus, the following applies to the temperature difference:

$$\Delta T = \frac{\dot{Q}}{\dot{m} \; c_{pW}}$$

By inserting the heat loss flow for the supply pipe in any section of the pipeline and under consideration of the mass flow rate, the temperature drop in the supply pipe in this section of the pipeline is as follows:

$$\Delta T_{cool} = \frac{\dot{Q}_L}{2 \; \dot{m} \; c_{pW}}$$

The following equation represents a more accurate calculation of the drop in the supply temperature in the function of distance for an earth-laid pipe section [44]:

$$T_{SPLY,i,1} = T_{gr} + (T_{SPLY,i,0} - T_{gr}) \; e^{-\frac{U_P \; 2 \; \pi \; r_P \; L}{c_{pW} \; \dot{m}}}$$

The supply temperature at the feed-in point of the section T_{SPLY_i0} is considered constant.

The temperature drop in the section of the supply pipe can be calculated as follows:

$$\Delta T_{cool} = T_{SPLY,i,0} - T_{SPLY,i,1}$$

$$\Delta T_{cool} = T_{SPLY,i,0} - \left[T_{gr} + (T_{SPLY,i,0} - T_{gr}) \; e^{-\frac{U_P \; 2 \; \pi \; r_P \; L}{c_{pW} \; \dot{m}}} \right]$$

The cooling of the heat carrier in the supply pipe for a certain transport distance is calculated as the sum of the individual temperature drops in the respective pipe sections as follows:

$$\Delta T_{cool,tot} = \sum \Delta T_{cool}$$

The supply temperature at the end of the respective pipe section is calculated as the difference between the supply temperature at the feed-in point and the pressure drop in the supply pipe for the entire transport distance as follows:

$$T_{SPLY,1} = T_{SPLY,0} - \Delta T_{cool,tot}$$

7.2 Pressure loss

7.2.1 Straight pipelines

The flow losses in pipe systems consist of the pressure losses in the straight pipeline sections and the sum of the individual losses due to pipe fittings such as elbows, junction pieces, cross section changes, equipment etc. [46].

The pressure loss due to pipe friction in straight pipe sections is calculated as follows:

$$\Delta p_L = \lambda \, \frac{l}{d_P} \, \rho_W \, \frac{v_m^2}{2}$$

The pressure loss calculation for pipe fittings (for more information see chapter 7.2.3):

$$\Delta p_L = \zeta \, \rho_W \, \frac{v_m^2}{2}$$

The relationship between the drag coefficient for pipe fittings ζ and the pipe-friction coefficient for pipe flow λ for a pipe section can be defined as:

$$\zeta = \lambda \, \frac{l}{d_P}$$

The Reynolds-number Re is defined as follows:

$$Re = \frac{v_m^2 \, d_P}{\upsilon} \; ; \; Re > 2'320 \rightarrow \text{turbulent flow}$$

Laminar convection becomes turbulent above a Reynolds number of 2320.

On the basis of common medium flow velocities v_m turbulent pipe flow can generally be assumed in district heating pipes. The pipe-friction coefficient for pipe flow λ therefore depends on the dimensionless Reynolds number and the pipe's roughness. Table 7.3 illustrates the pipe surface roughness k for different pipes and pipe materials. A reference value of ≤ 0.01 mm is recommended for district heating pipes [41].

The pipe-friction coefficient for pipe flow λ in one pipeline section can be read out with the Reynolds number Re, the inner pipe diameter d_P and the pipe's roughness k can be described in a Moody diagram (Figure 7.6) to ascertain the pipe-friction coefficient for pipe flow. The following equations are recommended to determine the pipe-friction coefficient. The more the laminar sub-layer covers the pipe surface roughness k, which acts as an additional turbulence generator, the lower the influence of k on the pipe-friction coefficient λ. The laminar sub-layer is therefore of great importance for the degree of turbulence and the pressure loss. There are three characteristic roughness ranges of turbulent flow [46].

However, empirical values show that with the pipe diameters usual in district heating and the maximum flow velocities according to [120], the pipe-friction coefficient λ can generally be assigned to the transition area (Area 2) (see also Figure 7.6).

Area 1: Hydraulically Smooth

The laminar sub-layer completely covers the pipe surface roughness k. The pipe flow, which in effect represents a turbulent interface, slides along the laminar sub-layer. Turbulence occurs automatically in the core flow (formation of vortices) and therefore only depends on the Reynolds number.

Validity Range:

$$2'320 < Re < \frac{d_P}{k} \log\left(0.1 \, \frac{d_P}{k}\right)$$

Approximate formula:

$$\lambda = \frac{0.309}{\left(\log \frac{Re}{7}\right)^2}$$

Area 2: Transition Area

With increasing speed, i.e. thinner sub-layers and/or increasing relative pipe surface roughness k, roughness peaks begin to emerge from the laminar sub-layer. The turbulence or the pipe-friction coefficient for pipe flow λ depends on k/d_P as well as on the Reynolds number.

Validity Range:

$$\frac{d_P}{k} \log\left(0.1 \, \frac{d_P}{k}\right) < Re < 400 \, \frac{d_P}{k} \log\left(3.715 \, \frac{d_P}{k}\right)$$

Approximate Formula:

$$\lambda = \frac{0.25}{\left[\log \left(\frac{15}{Re} + \frac{k}{3.715 \, d_P}\right)\right]^2}$$

Area 3: Hydraulically rough

With an even higher Reynolds number and/or higher k/dP, the roughness peaks emerge so far from the sub-layer that the energy losses (pressure loss) are caused only by the strong vortices (turbulences) emanating from this roughness accumulation. The pipe-friction coefficient for pipe flow λ depends only on k/d_P.

Validity Range:

$$Re > 400 \, \frac{d_P}{k} \log\left(3.715 \, \frac{d_P}{k}\right)$$

Approximate Formula:

$$\lambda = \frac{0.25}{\left(\log \frac{3.715 \, d_P}{k}\right)^2}$$

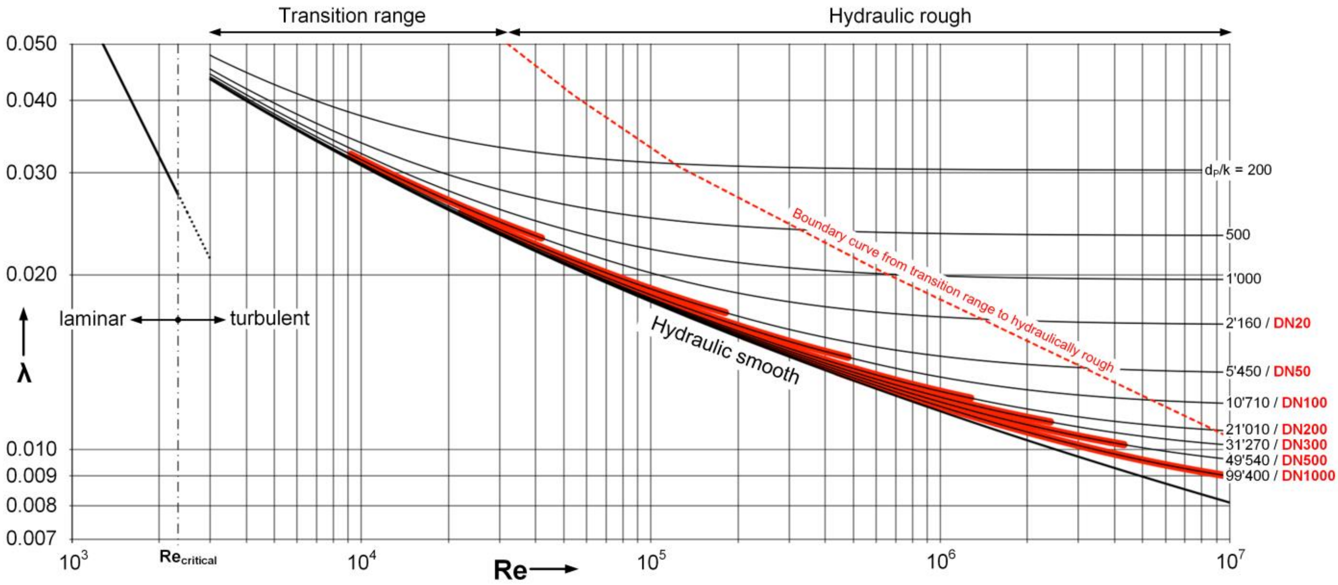


Figure 7.6 Moody diagram (Colebrook diagram) for the pipe-friction coefficient for pipe flow λ, based on the Reynolds number Re. Highlighted in red are common nominal widths and the corresponding range of the pipe-friction coefficient λ and the Reynolds number Re at a pipe surface roughness k of 0.01 mm and up to a maximum flow velocity according to [120].

Table 7.3 Pipe surface roughness k in mm for various pipes and pipe materials [49]

Material	Type	Condition	k-value
Copper Brass Bronze Light metal Glass	drawn or pressed	new (also steel medium pipes with given coating)	0.0013-0.0015
Rubber	pressure hose	new	0.0016
Plastic		new	0.0015-0.007
Steel	Seamless (commercial)	new - rolling skin - pickled - zinc-plated	 0.02-0.06 0.03-0.04 0.07-0.10
	longitudinal welded	new - rolling skin - bitumen-coated - galvanized	 0.04-0.10 0.01-0.05 0.008
	Seamless and longitudinal welded	used - moderately rusted or slightly encrusted	 0.1-0.2
		recommended **guided value** for the calculation of the pressure loss within district heating pipes [41]	**0.01**
Cast Iron		new - with cast-iron scale - bitumen-coated	 0.2-0.6 0.1-0.2
		used	0.5-1.5
Fiber-Cement		new	0.03-0.1
Concrete		new - carefully smoothed - striking off - center-line average - rough	 0.1-0.2 0.3-0.8 1-2 2-3

7.2.2 Pressure loss with corrugated pipes

In the laminar area, the pipe-friction coefficient for pipe flow λ for a corrugated pipe (Figure 7.7) is - similar to that of a smooth pipe - inversely proportional to the Reynolds-number. Information about λ-relationships in turbulent areas is rare to find in literature. Dependence on the Re number, in particular, is not backed up in the entire technically relevant area. The following is assumed for the validity range according to [45]:

Validity Range:

$$5 \times 10^4 < Re < 3 \times 10^5$$

$$0.2 < \frac{h}{a} < 0.6$$

$$0.0455 < \frac{h}{d} < 0.0635$$

Approximate formula:

$$\lambda = 3400 \left(\frac{h}{d}\right)^{4.13} \left(\frac{h}{a}\right)^{230 \left(\frac{h}{d}\right)^{2.1} - 0.7} Re^{0.193\, e^{\left[-3300 \left(\frac{h}{d}\right)^{2.6} \frac{h}{a}\right]}}$$

A simplified approach results in the following [45]:

Validity Range:

$$Re = 5 \times 10^4$$

$$0.2 < \frac{h}{a} < 1.2$$

Approximate formula:

$$\lambda = 0.2 \left(\frac{h}{d}\right)^{0.6} \left(\frac{a}{h}\right)^{0.7}$$

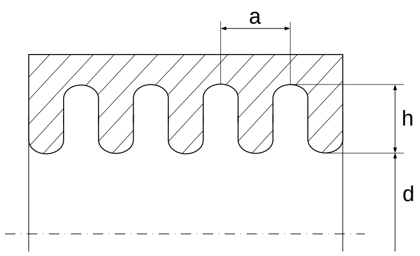


Figure 7.7 Cross-section of a corrugated pipe with the necessary dimensions to calculate pipe resistance.

An increased pipe-friction coefficient λ at Re > 4 x 10^4 is typical for the turbulent flow in corrugated pipes. This phenomenon is attributed to the formation of different vortex systems (primary and secondary vortex) in the wall protuberances. Corrugated pipes are usually threaded. The medium is therefore influenced by both axial and rotational movements. However, this influence on resistance behavior is negligible. The rotational intensity, on the other hand, is important for heat transfer, as it can prevent "dead water zones" in the grooves [45].

With identical Reynolds numbers, corrugated pipes have a far greater pipe-friction coefficient λ (factor 2 to 15 times greater). To ensure that the specific pressure loss per routed pipeline meter is not too high, corrugated pipes usually have to be designed one to two nominal diameters larger than smooth pipes.

7.2.3 Pressure loss of pipe fittings

Fittings such as manifolds, junctions, valves, etc. cause additional pressure loss. Basically, after installation of fittings, pipe flow is disturbed on a length of 10 to 30 times the length of the diameter by turbulent flow, whereby further pressure is lost in addition to the loss in valves, heat exchangers, etc. in the subsequent pipeline, compared to pipe loss in straight pipes.

The ζ value of fittings includes all losses resulting from their installation as compared with losses in straight pipelines.

Derivations and lists of the many different calculation models for pipeline installations are not gone into here. The following books are recommended for estimations, as they contain information about the loss coefficients of the most commonly used fittings: Glück [45], Wagner [49] and Böswirth [46] as well as in English by Idelchik [47]. The manufacturer's specifications for the respective fittings should also be used for further planning.

7.3 Dimensioning pipe diameter

The pipe diameters are dimensioned on the basis of the specific pressure loss per meter of pipe length. For this purpose, guide values of specific pressure losses are stated.

“QM Holzheizwerke” [21] recommends designing for a medium specific pressure drop of between 150 Pa/m and 200 Pa/m for a critical node. In most cases, the critical node corresponds to the district heating line from the heat station to the most distant customer in the grid.

Due to the varying nominal diameters of the district heating pipes, it is not possible to achieve an average pressure loss of 200 Pa/m in a longer pipe section with changing pipe diameters. The individual sections, if designed consistently, will have a maximum specific pressure loss of e.g. 200 Pa/m. For this reason, and based on practical experience, it is advisable to design the individual pipe sections to a maximum specific pressure drop of between 250 Pa/m and 300 Pa/m for the planned final capacity stage at design temperature. The maximum specific pressure drop is generally only achieved during very few operating hours, the rest of the year the district heating network is so to speak "overdesigned".

Figure 1.4 shows recommendations for nominal diameters compared to maximum flow rates with constant specific pressure drops of 100 Pa/m, 200 Pa/m, and 300 Pa/m. The maximum flow velocities with constant specific pressure losses are calculated assuming pipe friction coefficients of 0.020 for DN 20, 0.016 for DN 80 and 0.015 for DN 400. The pipe friction coefficients are based on the approximation formula for the transition area (Chapter 7.2.1). The comparison shows that the recommendations according to “ÖKL” [120] for main and branch lines up to DN150 lead to a similar interpretation as the consideration of maximum pressure losses at a little less than 300 Pa/m.

When dimensioning and calculating pipe diameters, a hydraulic pipe surface roughness of typically 0.04 up to 0.05 mm is often used. However, according to [41], a maximum hydraulic pipe surface roughness of 0.01 mm is recommended for calculating the pressure loss.

7.3.1 Recommendations

For dimensioning the pipe diameter, the following recommendations apply:

- Hydraulic system **pipe surface roughness** k ≤ 0.01 mm
- Design of the individual pipe sections for the planned final capacity stage at design temperature to a maximum **specific pressure drop** of between **250 Pa/m and 300 Pa/m**. The nominal diameter at which the maximum specific pressure drop is reached should be selected.
- Use an **average specific pressure drop** of between **150 Pa/m and 200 Pa/m** (e.g. for the critical node) as a control variable.

However, when sizing and assessing pipe diameters, there are other factors that are difficult to estimate during planning, such as:

- Effective temperature difference during operation
- Effective power requirement at final capacity stage
- Estimation of extension reserves
- Estimated renovation potential of the building estate to be connected for the planned life cycle of the district heating network at the time of planning.

For pipe systems consisting of pre-insulated rigid and flexible steel pipes (also duo), and pre-insulated PEX pipes (also duo), Figure 13.1 presents reference values for possible transfer rates at different temperature differences of 15 K, 30 K and 45 K with a specific pressure loss of 300 Pa/m. Figure 13.2 indicates values for possible transmission rates at different specific pressure losses of 100 Pa/m, 200 Pa/m and 300 Pa/m at a temperature difference of 30 K.

The values are based on the following assumptions:

- Density 983 kg/m^3
- Kinematic viscosity 4.74 10^{-7} m^2/s
- Heat Capacity 4.183 kJ/(kg K)
- Pipe surface roughness 0.01 mm

7.3.2 Procedure

7.3.2.1 Preliminary

First, a pipeline plan must be drawn up, i.e. a schematic, non-scale representation of the complete heating network. The pipeline plan serves as the basis for the design of the heating network. The pipelines are structured according to:

- Main pipeline(s)
- Branch pipelines
- House connection pipelines

The pipeline plan is then divided into sections, whereby a section must have the same flow rate and pipe diameter. In the course of the calculation, it records:

- Number of the pipe section
- Heat load of the section
- Length of the section
- Total ζ value of pipe fittings in the section
- Pipe diameter of the section.

Regardless of the calculation method, a section is calculated as follows:

1. Calculation of the flow rate based on heat load and temperature difference
2. Determination of the length of the section (supply + return)
3. Determination of the sum of the individual resistances within the section of the pipeline
4. Preliminary determination of the pipe diameter based on heat load and temperature difference based on Figure 13.1 and Figure 13.2.
5. Calculation of the actual flow velocity
6. Calculation of the actual specific pressure drop.

7.3.2.2 Design in 4 Steps

The actual design of the heating network is carried out in four steps. One section at a time is calculated – if necessary, several times. Since the heating network as a whole can only be designed iteratively, the procedure is as follows:

1. Provisional calculation of the most unfavorable route
2. Recalculation of the most disadvantageous route
3. Preliminary calculation of the remaining branch and house connection pipelines
4. Recalculation of the entire heating network.

Step 1:
Preliminary determination of the most unfavorable route (usually the most distant customer) consisting of

- the most unfavorable main pipeline
- the most unfavorable branch pipeline
- and the most unfavorable house connection pipeline.

Determining a provisional pressure difference based on the most unfavorable consumer (a relatively high value may be necessary for sufficient valve authority). Preliminary determination of the pipe diameter based on heat output and temperature difference presented in Figure 13.1 and Figure 13.2.

For the main pipeline, a lower pressure loss can initially be assumed than for the branch pipe. This results in a more favorable hydraulic system behavior due to the branch lines' higher authority.

If the sum of the Zeta values of the individual resistances is not already known, 10% to 20% can be added to the pipe length.

Step 2:
A recalculation of the pressure losses taking actual pipe diameters into account, and the final Zeta values of the individual resistances. The result is the actual pressure differences at the bifurcations that can be used to design the remaining branch pipes.

Step 3:
The procedure is basically the same as in step 1. However, the pressure difference at the bifurcation is specified here, which must be "used up" in the respective branch and its least favourable house connection pipe.

Step 4:
Finally, the most disadvantageous route is determined. This may have changed from the assumption in step 1. The pressure losses are recalculated based on the definitive pipe diameters and the Zeta values of the individual resistances. Where enough large pressure differences are available, smaller pipe diameters can be selected.

7.3.3 Calculation Methods

Pressure loss can be calculated using different methods for the purpose of dimensioning the pipeline.

7.3.3.1 Manually

Using tables and pipe network calculation forms. This requires:

- R value, flow rate, velocity, stagnation pressure (dynamic pressure) for different pipe diameters
- Zeta values for different components

As stated in Chapter 7.3.1, using a maximum hydraulic system pipe surface roughness of 0.01 mm is recommended. Since publications available today are calculated with a considerably higher pipe surface roughness. An R-value table with a hydraulic system pipe surface roughness of max. 0.01 mm can be found in Appendix 13.3.

7.3.3.2 Simulation Software

There are pipeline network calculation programs solely used to calculate pressure losses, for example on the internet the free pipeline network calculation program provided by the "Zentrum für Integrale Gebäudetechnik ZIG" of the Applied University of Lucerne (": www.hslu.ch/zig). In this program, the pipe surface roughness can be freely specified.

There are also more extensive calculation programs, which in addition to the pressure loss calculation also include heat loss determination, thus enabling optimisation of the entire system. Commercial calculation programs are, for example, the following:

- EC-Netz
- STANET
- R-Design
- ROKA GS
- SIR-3S
- Termis

For calculations, it should be noted that a pipe surface roughness > 0.01 mm (approx. between 0.04 and 0.05 mm) is usually pre-set. This value can usually be adjusted.

7.4 Pump design

7.4.1 Characteristic pump curve

The characteristic pump curve shows the delivery head in the function of the pump capacity. A distinction can be made between flat and steep pump characteristic curves (Figure 7.8). The steeper the characteristic pump curve, the greater the pressure fluctuations and the disturbance in the hydraulic network when the volume flow changes. However, a steep characteristic curve is advantageous for the control of the pressure difference.

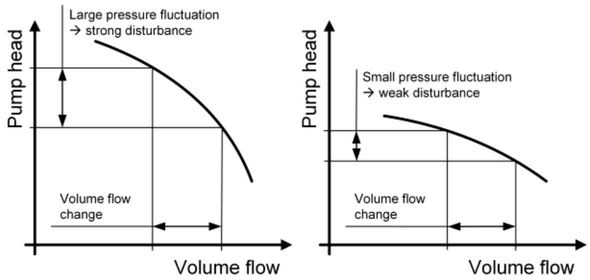


Figure 7.8 Characteristic pump curve: Steep (left) and flat (right) [21].

7.4.2 Characteristic system curve

The pressure losses of the grid increase with the volume flow, as represented in the characteristic system curve. Since both the characteristic pump curve and the characteristic system curve describe the interaction between pressure (delivery head) and volume flow, both can be entered into the same coordinate system (Figure 7.9). Both characteristic curves have a common intersection point. This is the operating point of the pump. At the operating point, the delivery head equals the pressure loss of the system.

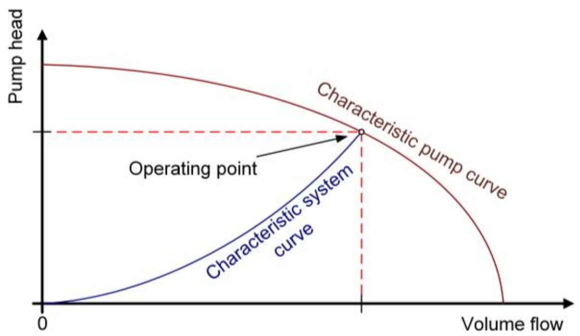


Figure 7.9 Characteristic pump and system curve in the same coordinate system.

7.4.3 Proportionality laws

When changing the speed of a circulating pump, the delivery head, volume flow and hydraulic performance follow the three laws of proportion.

The volume flow is proportional to the rotational speed of the pump:

$$\frac{\dot{V}_1}{\dot{V}_2} = \frac{n_1}{n_2}$$

The pump head (pressure difference) changes with the square of the speed:

$$\frac{H_1}{H_2} = \frac{\Delta p_1}{\Delta p_2} = \left(\frac{n_1}{n_2}\right)^2$$

The hydraulic pump performance varies with the cube of the speed:

$$\frac{P_{hydr\,1}}{P_{hydr\,2}} = \left(\frac{n_1}{n_2}\right)^3$$

At half the speed, the volume flow drops to half, the pump head, i.e. the pressure drop, falls to a quarter and the pump's power demand to an eighth.

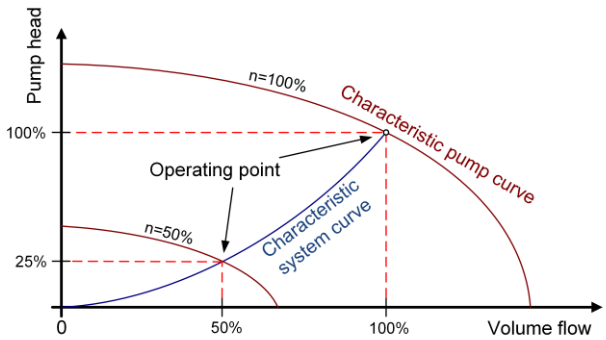


Figure 7.10 Characteristic pump and system curve for two different rotational speeds.

7.4.4 Required pump capacity

The volume flow required of the district heating pump depends on the flow rate in the entire heating network, i.e. the required flow rate in the first section immediately after the district heating pump. The volume flow relevant for the design of the pump equals the heat demand of the customers and the heat losses to be compensated for in the network on the coldest day of the year at a defined temperature difference in the network.

$$\dot{V}_P = \frac{\dot{Q}_N}{\rho_W \; c_{pW} \; \Delta T_N}$$

The required pump head of the district heating pump is calculated based on:

- Pressure loss to the most unfavorbarly situated customer (unfavorable main, branch and house connection pipeline → critical node CN)
- Required pressure difference above the house service connection of the least favourable customer
- Factors hitherto not taken into account such as pressure losses before the district heating pump, like the control valve of the pre-regulator, storage, connecting pipes, etc., which are located mostly in the heat station.

$$H_P = \frac{\Delta p_{CN} + \Delta p_{HC} + \Delta p_{HGR}}{\rho_W \; g} = \frac{\Delta p_P}{\rho_W \; g}$$

Thus, the operating point is known under design conditions and district heating pumps with the suitable pump characteristic curve can be chosen accordingly. Since pump efficiency (hydraulic system and electrical efficiency) is highly dependent on the pump capacity, care must be taken during design that the pump reaches its capacity and the delivery head at the outlet in the most frequently operating point (usually at partial load) has a high efficiency. The hydraulic system efficiency data are determined by the manufacturers by measurements on the test bench and are presented in the characteristic pump curve. The highest efficiency for district heating pumps usually ranges between 65% and 85%. For an estimation, an efficiency (hydraulic system and electrical) of 75% can be applied.

The pump performance can be calculated from the pump capacity and the delivery head as follows:

$$P_P = \frac{\Delta p_P \; \dot{V}_P}{\eta_{hydr} \; \eta_{el}} = \frac{H_P \; \rho_W \; g \; \dot{V}_P}{\eta_{hydr} \; \eta_{el}} = \frac{H_P \; \rho_W \; g \; \dot{V}_P}{\eta_P}$$

7.4.5 Pump energy demand

It is important that pumps are used with the highest possible degree of efficiency. Normally, the efficiency of the pump must be calculated at the operating point. That it makes sense to compare pumps with each other is demonstrated in Figure 7.11. Smaller pumps tend to have lower efficiencies than larger ones. Small pumps, for example, hardly achieve more than 50 % even at the best operating point. An essential improvement, even with small pumps, has recently been achieved by using DC synchronous motors with permanent magnet rotors.

The annual energy requirement of the pumps is determined by the summed average values of the resulting flow rates and delivery pressures from the annual duration curve of the network. The operating mode of the pumps and the pump-specific efficiency has an influence on the power requirement. As an estimated approximation of the pump power demand, the following reference is sufficient, in which the nominal pump output is multiplied by the full-load operating hours of the customers:

$$E_P = P_P \; \tau_{HC}$$

From experience, the annual energy requirement of the pumps is between 0.5 % and 1.0 % of the distributed heat, at an optimum design of the district heating network.

7.4.6 Design specifications

District heating pumps are the heart of district heating systems. Without their safe functioning, heat delivery is not possible. This section deals with specific features in the design and configuration of pumps in district heating systems in order to ensure supply even in the event of changing operating conditions. In the case of district heating networks, this becomes clear during commissioning when operation is started with a small number of customers. Often network dimensioning planned for later expansion has already been realised. This means that operation is started with low pump capacities and significantly lower differential pressures in the network.

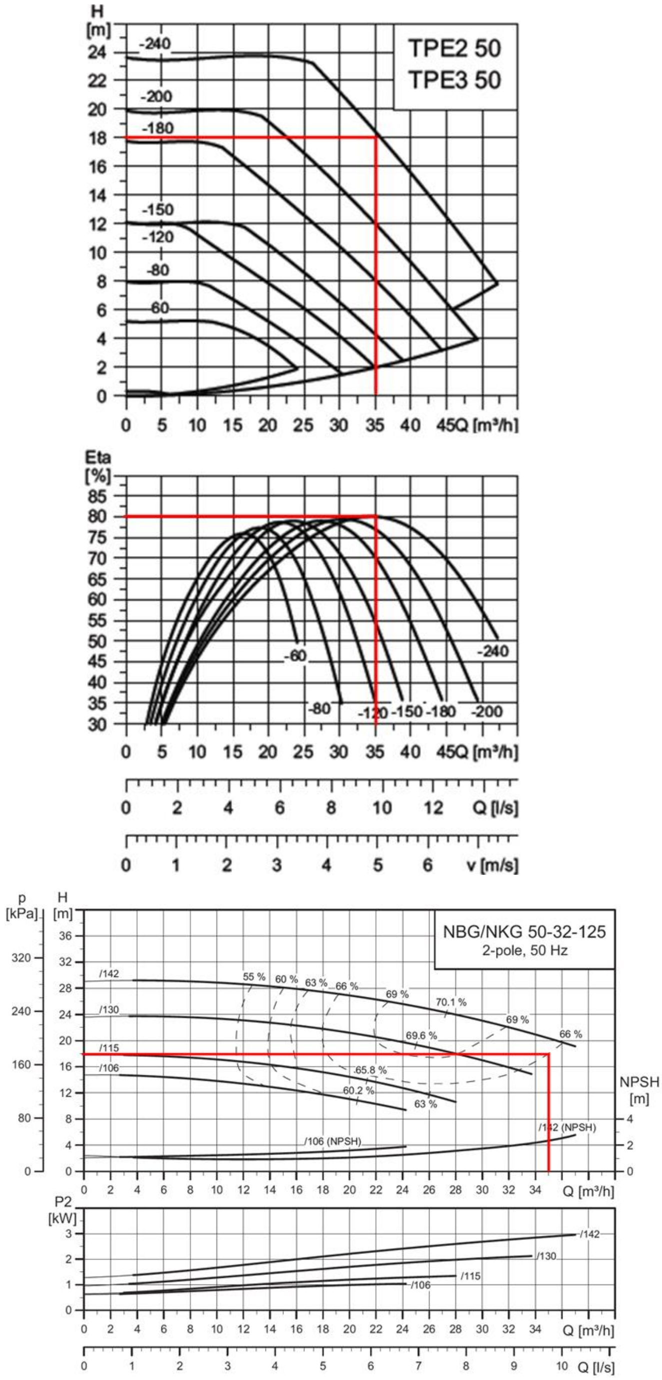


Figure 7.11 Example of two pumps with a delivery head of 18 m and a pump capacity of 35m³/h [source Grundfos]. Depending on the design and the position of the operating point, there are different efficiencies: 80 % (above): inline pump with impeller diameter 240 mm and highly efficient permanent magnet motor. 66% (bottom): Standard pump with impeller diameter 125 mm with standard motor IE3 (asynchronous motor).

7.4.6.1 Problem of Over-current

As mentioned in the previous section, pumps should be operated in the range of high efficiency in typical operating conditions. However, it should be taken into account that, in the case of low-capacity or the network not yet in the planned final capacity stage, the required pump pressure is low. In this case, due to low hydraulic grid resistance and at a specifically high pump capacity, the pump flow can increase and exceed the limit of the over-current protection of the pump motor. In order to make pump operation more robust, it is recommended to select the operating point (characteristic system curve) of the pump at a significant distance from the maximum pump capacity Q_{Max}. This can prevent the motor's current consumption going over the limit. A slightly reduced pump efficiency may have to be accepted for this reason. The excess current limit must also be taken into account in the partial load case, as the motor will consume less current and be slower (Figure 7.12).

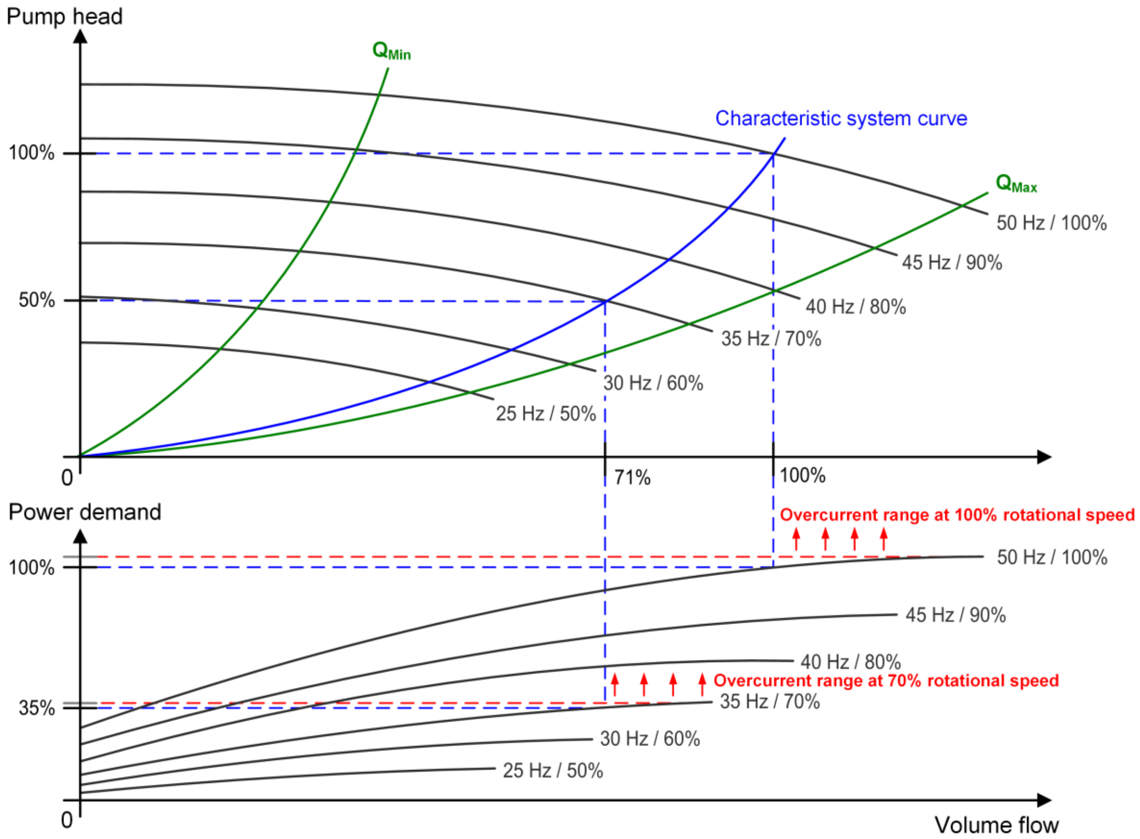


Figure 7.12 Example characteristic pump curve for a speed-controlled pump. The delivery head (above) and the power consumption (below) depending on the pump capacity as well as the representation of the maximum (Q_{Max}) and minimum (Q_{Min}) permissible capacity of the pump (above) are presented. In addition, a system characteristic curve (blue) and the area of the over- current (red) are entered as examples.

7.4.6.2 Pump flow rate

Cavitation can be avoided if there is sufficient pressure build-up in district heating pumps. However, damage due to cavitation and erosion can occur at high flow velocities, especially at pump inlets and pump outlets. It is therefore advisable to design for robust pump operation by selecting inlet cross-sections and exit cross-sections in such a way that under normal operating conditions no particularly high inlet speeds (> 3 m/sec. for DN100) and exit speeds occur. Hence, during the pump planning and design process the inlet and outlet cross-sections should be selected accordingly so that the pumps can cope with higher pump capacities if necessary.

7.4.6.3 Minimum Pump Capacity

Pump manufacturers specify performance-dependent minimum pump capacities QMin for the operating mode of the pumps (Figure 7.12). These are 25% of the maximum pump capacity for typical district heating pumps. The minimum pump capacity serves on the one hand to protect the pump from overheating, on the other hand it serves to protect the sliding ring seals, which rely on lubrication and cooling by the system medium. Dropping below the minimum pump capacity should be avoided and ensured against while taking into account the medium temperature, e.g. by a system bypass when the network volumetric flow rate is low.

7.5 Pipe statics

This section contains calculation bases, indications and design features for strength assessment and the correct design of pipes and pipe components. It shows basic correlations and methods for calculating the pipe strength and for the calculation of heat-related stresses, as well as some aids and diagrams for the design of pipe brackets and expansion compensation. Aspects of laying technology are discussed for evaluating the static design of underground lines.

A comprehensive static calculation and the static proof of district heating lines, in particular for the most frequently used underground pipe systems, cannot be discussed in this handbook for reasons of space. For this purpose, explicit reference is made to relevant documents such as AGFW leaflets (FW 401 [100], FW 410 [101] and FW 411 [102]) or to standards such as SN EN13941 [94].

7.5.1 What is pipe static?

The purpose of calculating and documenting pipe statics is to be able to install, insulate and operate heat distribution pipelines in such a way that no damage is caused by movement over an appropriately long service life (at least 50 years for earth-laid and at least 80 years for open-laid systems). The plant operator is responsible for the proper execution and the correct operation of pressure devices, including pipes with district heating media. It is the operator's responsibility to demand static proof during the construction of a district heating system.

Knowing how to handle the longitudinal expansion of pipes directly affects route planning (for more information see chapter 4.7.2). The planner should be aware of the following issues:

- What happens to the pipe during heating?
- How can longitudinal expansion be compensated?
- What defines prestress and what are the resulting effects?

A pipe expands in both directions when heated from a fixed point; this expansion must be compensated in some way (Figure 7.13).

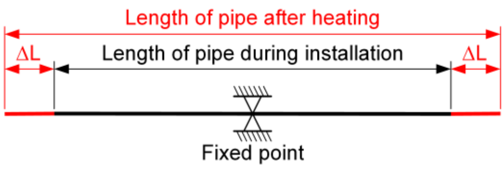


Figure 7.13 Schematic representation of the thermal expansion.

The **longitudinal expansion** is absorbed naturally via the extension limbs (L-, Z- or U-arc), and compensators can also be used for overhead lines.

The **prestress** reduces the length expansion of the pipes under operating conditions. This can, for example, reduce extension limbs. A distinction is made between thermal and a mechanical (cold) prestress.

The following chapters provide an insight into the physical basics as well as solutions for the installation and serve the technician, planner and engineer to be able to imagine basic pipe static relationships and to answer simple questions about the pipe static calculation themselves.

Note: The explanations on pipe static calculations refer to the steel grades P235TR1, P235TR2 and P235GH (St. 37.0 and St. 35.8, respectively) commonly used in district heating pipeline construction. For PE or PEX medium pipes, despite about 20 times the thermal expansion coefficient, completely different conditions apply and are dealt with at the end of the chapter.

7.5.1.1 Pressure resistance and wall thickness

The following section is devoted to the simplified calculation of the stresses occurring solely by the pressure in vessels or pipes and the resulting wall thicknesses. This simplified calculation already allows an evaluation or determination of stresses and wall thicknesses with mostly reliable results for the design or rough design of a vessel or pipe.

The pressure resistance and wall thickness of cylindrical vessels and pipes are calculated by looking at the multi-axis stress state. In the case of a cylindrical shape, this consists of the overlay of longitudinal, circumferential, and radial or internal pressure stresses. Derived from strength hypotheses, the calculation of equivalent tensile stresses is carried out in a simplified form. This reduces calculating complex stress states to easily applicable relationships.

The strength analysis is carried out according to the diagram is shown in Figure 7.14. The left branch can be summarized under the term stress analysis, while the right branch presents the load-capacity analysis. The two boundary conditions are linked under the requirement that the load in the component does not exceed the load-bearing capacity of the material [50].

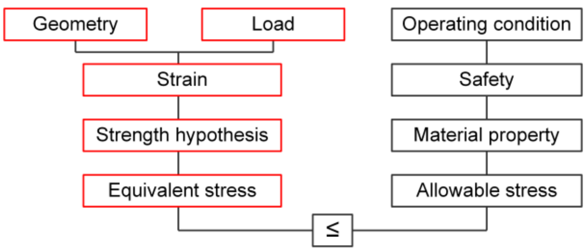


Figure 7.14 Calculation scheme for strength analysis according to [50]. On the left: stress analysis; on the right: load-capacity analysis.

The strength condition is expressed in the following equation according to Figure 7.14 by comparing the equivalent tensile stress σ_V according to the shear stress hypothesis (SH) with the permissible stress σ_{adm} in the component:

$$\sigma_V(SH) \leq \sigma_{acc}$$

The equivalent tensile stress σ_V according to the shear stress hypothesis (SH) for pipes and cylindrical vessels, also called boiler formula, is expressed as:

$$\sigma_V = \frac{p\ d_m}{2\ s}$$

The permissible stress must ensure that the calculated stress remains below the corresponding strength parameter of the material (R_e, R_m) with sufficient safety S.

The permissible stress must ensure that the calculated stress remains below the corresponding strength parameter of the material (YTS, UTS) with a sufficient safety S.

$$\sigma_{acc} = \frac{YTS}{S_{YTS}} = \frac{UTS}{S_{UTS}}$$

Standard safety factors range by:

- $S_{YTS} = 1.5$
- $S_{UTS} = 2.4$

The wall thickness s can be calculated by converting the boiler formula as follows:

$$s = \frac{p\ d_m}{2\ \sigma_V}$$

The calculated pipe wall thickness sV can also be calculated according to Table 7.4.

When determining the required wall thickness, the manufacturing tolerances and the wear caused by corrosion must be taken into account by appropriate allowances (c_1, c_2).

The minimum wall thickness s is thus:

$$s = s_V + c_1 + c_2$$

The wall thickness s_e to be ordered is therefore:

$$s_e \geq s$$

More elaborate specifications and additional information regarding safety allowances can be found in [50] "Wagner W. Festigkeitsberechnungen im Apparate- und Rohrleitungsbau" (8th Edition 2012, Page 33 ff).

Table 7.4 Equations for calculating the pipe or cylinder wall thickness sV under internal pressure without allowances and tolerances with predominantly static loading (based on [50]).

Calculation Instruction	Application Limits	Material stress and equivalent tensile stress	Calculation equations with	
			Outer Diameter	Inner Diameter
SN EN 13480-3 [93]	$d_a/d_i \leq 1.7$	fully plastic shear stress hypothesis (medium stress)	$s_V = \frac{d_a\ p}{2\ \sigma_{acc}\ \upsilon_N + p}$	$s_V = \frac{d_i\ p}{2\ \sigma_{acc}\ \upsilon_N - p}$
	$d_a/d_i > 1.7$	Lamé-function	$s_V = \frac{d_a}{2}\left(1 - \sqrt{\frac{\sigma_{acc}\ \upsilon_N - p}{\sigma_{acc}\ \upsilon_N + p}}\right)$	$s_V = \frac{d_i}{2}\left(\sqrt{\frac{\sigma_{acc}\ \upsilon_N + p}{\sigma_{acc}\ \upsilon_N - p}} - 1\right)$
AD2000 [119]	$d_a/d_i \leq 1.2$ (für da ≤ 200 mm $d_a/d_i \leq 1.7$)	fully plastic shear stress hypothesis (medium stress)	$s_V = \frac{d_a\ p}{2\ \sigma_{acc}\ \upsilon_N + p}$	$s_V = \frac{d_i\ p}{2\ \sigma_{acc}\ \upsilon_N - p}$
	$1.2 \leq d_a/d_i \leq 1.5$	elastic GE-Hypothesis (Peak stress)	$s_V = \frac{d_a\ p}{2.3\ \sigma_{acc} - p}$	$s_V = \frac{d_i\ p}{2.3\ \sigma_{acc} - 3\ p}$

ν_N = Weld seam quality (for seamless pipes =1, for welding seams depending on the execution 0.7-1)

The result of pipe or cylinder wall stress calculation is only valid if material stress is caused by pressure and if the permissible stresses of the respective material are taken into account. These permissible stresses depend, among other things, on the minimum and maximum operating temperature and the wall thickness. Thus, the permissible tension in metallic materials decreases with increasing temperatures compared to ambient temperatures and with increasing wall thicknesses. A typical piping material P235GH (material number 1.0345, former designation St35.8 or "Kesselblech HII") has a permitted stress at room temperature of 235 N/mm^2 for a wall thicknesses of < 16 mm. For wall thicknesses that range from 40 to 60 mm, the permissible stress at room temperature is only 215 N/mm^2. At a temperature of 100 °C, the permitted stress is still at 198 N/mm^2 and for 200 °C it ranges still by 170 N/mm^2.

7.5.1.2 Bending stress

The calculation of the bending stress σ_B for pipes is carried out by the following correlation:

$$\sigma_B = \frac{T_B}{W_B}$$

The bending moment M_B must be assessed and calculated individually depending on the load. The section modulus W_B for a hollow shaft or pipe is calculated as follows:

$$W_B = \frac{\pi\ \left(d_a^4 - d_i^4\right)}{32\ d_a}$$

7.5.1.3 Thermal expansion and heat stress

A component clamped on one side as presented by Figure 7.15 expands by the amount ΔL when heated from the initial temperature T_0 to the final temperature T_1.

$$\Delta L = L\ \alpha\ \Delta T$$

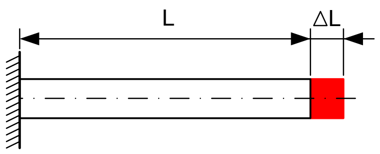


Figure 7.15 Linear extensional strain of a component clamped on one side (based on [50]).

If thermal expansion is obstructed, tensions arise (Figure 7.16). The cause of the obstruction can be clamping, different temperatures in different places in the component or uneven expansion coefficients in the component.

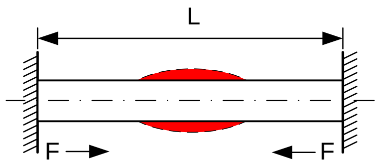


Figure 7.16 Linear extensional strain of a component clamped on both sides (based on [50]).

The linear compression or expansion is defined as:

$$\varepsilon = \frac{\Delta L}{L}$$

The stress in the area of Hooks law (elastic range) is proportional to the strain or linear compression. The following relationship results with the proportionality factor E (modulus of elasticity):

$$\sigma = \varepsilon\ E$$

The maximum permissible values of the tension result from the material used for the component and the temperatures prevailing in the application. The stress generated during use at maximum temperature difference must not exceed the maximum permissible stress.

$$\sigma_T < \sigma_{acc}$$

The stress at maximum temperature difference is determined by the following:

$$\sigma_T = \varepsilon\ E = \frac{\Delta L}{L}\ E = E\ \alpha\ \Delta T$$

The maximum temperature difference is hereby:

$$\Delta T_{acc} = \frac{\sigma_{acc}}{E\ \alpha}$$

Table 7.5 The Coefficient of linear expansion α_L and the E-Module of some materials.

Material	Coefficient of linear expansion 10^{-6} / K	E-Module 10^3 N/mm²
Unalloyed steel	12.0	210
High-alloyed steel	10.0 (ferrous), 16.7 (austenitic)	200
copper	16.5	100-130
Aluminum	23.0	70
grey cast iron	9.0	90-145

7.5.1.4 Overlay of stress

For a pipe-static design and static proof, all acting stresses and their overlays must be included in the calculation. It is therefore not sufficient to design a pipe wall or container wall solely on the strength due to the pressure, but all superimposed stresses of bending, torsion, buckling, heat and their reciprocal effects must be integrated into the static proof. For the calculation and evaluation of other stress types such as buckling, torsional, shear and heat stress, recommendations are made in [50].

In addition to the determined stresses under maximum load conditions, the temporal distribution of the load is added for an evaluation of the statics. The load on a component (pipe) can be static (resting) or dynamic (alternating). A material usually has a higher fatigue strength when a static load remains constant (e.g. constant internal pressure in a vessel), than when the load changes (pressure fluctuations) and when the load swings (change tensile stress - compressive stress). A high load change rate leads to a reduction in resistance.

Especially in the case of district heating pipelines and systems, temperature fluctuations that correspond with load changes can be expected. These load changes in a heating or cooling process vary between the minimum and maximum operating temperature. For the static calculation of a district heating pipeline, the load change number must be included as the basis for calculation. For example, pipelines that are shut down according to the season have significantly higher load changes than pipelines that have only minor performance-related temperature fluctuations.

7.5.1.5 Static proof

All overlapping mechanical stresses are included in the consideration for static proof. Therefore, static proof can be very complex. Calculation programs such as SISKMR, ROHR2, CAESAR, AUTOPIPE and EASYPIPE, enable extensive static proof depending on the application. Due to the complexity and the required quality of pipe static calculations, experience using the application for targeted processing is helpful. Thus, the static proof is often provided by institutions and persons who bring the appropriate knowledge and experience in resistance theory and statics.

- The result of the static calculations for pipelines includes the following points:
- Definition of the regarded pipeline section

- Reference to the suitability of the chosen material.
- Proof of compliance with material-induced stress limits for the considered pipe section in the context of operational temperatures.
- Detection limit compliance, taking into account the estimated number of load changes
- Proof of the statics of the pipe mounting bracket
- Indication of the triaxle stresses and torques for all elements of the considered pipe segment.
- Static proof of the pipe support (anchor points, floating bearing).

For earth-laid pre-insulated rigid steel pipes the following information is added (find additional information in chapter 7.5.3):

- Loading of the insulating material (PUR foam) and the PE coating.
- Suitability and dimensioning of compensation elements such as expansion pads.

The results of the static calculations may lead to the following measures in the event of design value violations:

- Adjust wall thicknesses of pipe elements, bends, T-pieces, bores.
- Reduce overloading of a pipe element by adapting the geometric arrangement.
- Adjust positioning of anchor points and other clamping points.

After carrying out measures, re-checking the statics in corrected arrangement and verifying if design values are in a suitable range, the static suitability of the pipe section can be determined.

7.5.2 Laying technique and specification

7.5.2.1 Expansion compensation

Natural pipe expansion compensation is usually preferred rather than using compensators, especially if the routing has a geometry that allows natural expansion compensation anyway. It is important that the pipes are laid or mounted with the lowest possible stress after mounting.

In the case of earth-laid and free-laid pipes, natural expansion compensation is made possible by changing direction of the pipelines. The resulting extension limbs facilitate the elastic bending of the pipe, thus compensating for the changes in length in a natural way. Extension limbs or strain equalizers are realised as angular arc (L-arc), double angle arc (Z-arc) and (U-arc) (Figure 7.18).

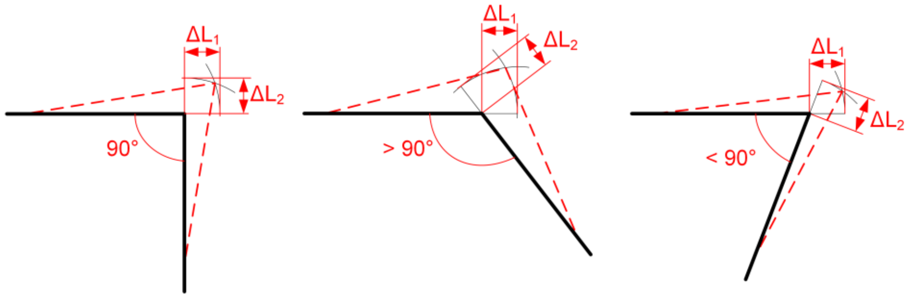


Figure 7.17 Exemplary representation of the angular position of an L-arch. The thermally caused expansions ΔL ($\Delta L_1 = \Delta L_2$) are identical in these three cases.

The specific conditions for earth laid pre-insulated rigid pipes are described in chapter 7.5.3, for free laid pipes in chapter 7.5.4 and for pre-insulated PEX pipes in chapter 7.5.5.

The angular position of the extension bend has an additional influence on the expansion range, which must be compensated by expansion pads (Figure 7.17). If possible, the arches must be executed at right angles, as the expansion can best be compensated with 90° arcs. For arcs with an obtuse angle (> 90°), the expansion range increases compared to a 90° arc. Arches with too acute angles (< 90°) are generally not recommended.

The effect of the expansion is mitigated by thermal prestress. In high-temperature systems, it is essential to prestress the pipelines. Find additional information about pre-stress techniques in the chapters 7.5.3 and 7.5.4.

7.5.2.2 Pipe static design temperature

The pipe static design temperature ΔT_{PSD} is always decisive for the calculation of the pipe static calculation. This can be calculated as follows from the design temperature T_{Des} and the laying temperature T_{Lay} :

$$T_{PSD} = T_{Des} - T_{Lay}$$

For earth laid pipe systems, a design temperature T_{Des} of 110 °C is recommended at a maximum operating temperature below 100 °C. At a maximum operating temperature above 100 °C, a design temperature T_{Des} of 130 °C, but at least 10 °C above the operating temperature, is recommended.

For freely laid pipe systems, a design temperature is recommended depending on the operating temperature. The design temperature T_{Off} is 10 °C higher than the maximum operating temperature.

The following values can be assumed for the laying temperature or the outside temperature, when laying the pipes:

- Earth laid: Summer: 20 °C
  Spring/Fall: 10 °C
  Winter: 0 °C
- Exposed Piping in basements, garages and buildings: generally 20 °C

Note: There is no distinction between the supply and return temperature! The return is treated in the same way as the supply during static observation.

7.5.3 Earth-laid pipes

Due to the proven technology and long service life, pre-insulated rigid steel pipes are the most widely used pipe system for earth-laid district heating pipelines.

As mentioned above, natural expansion compensation is realised by changing direction of the pipelines using an angle elbow (L-arc), double angle elbow (Z-arc) and U-arc. Assuming the leg length L_L (side length) of the types of expansion compensation presented in Figure 7.18 is identical for all three types, the L-arc shows the smallest and the U-arc shows the greatest flexibility for expansion compensation.

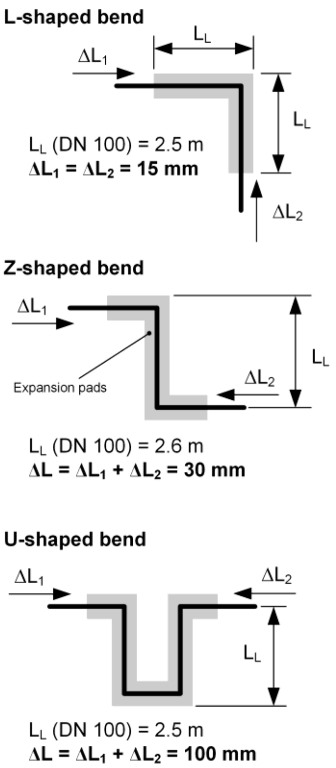


Figure 7.18 The example presents a comparison of the maximum possible expansion absorption with identical leg lengths L_L (side lengths) according to Brugg Pipe Systems ("Premant DN 100").

This is shown by the installation of a pre-insulated rigid steel pipe DN 100 according to Brugg Pipe Systems (the "Premant DN 100"): With an identical leg length L_L of 2.4 m or 2.5 m, the maximum compensatory pipe expansion ΔL is approximately 15 mm, when laying an L-arc (without anchor points); with a Z-arc a total of 30 mm can be compensated ($\Delta L_1 + \Delta L_2$) and with a U-bow a total of 100 mm can be compensated ($\Delta L_1 + \Delta L_2$). Thus, a U-bow for expansion compensation for pre-insulated rigid steel pipes shows the highest flexibility.

For the assessment of the extension limb and the necessary expansion pad lengths, reference is made to the designs and diagrams (parts 10 and 11] presented in the "AGFW Arbeitsblatt FW 4012 [100] as well as to the manufacturer-specific information.

It should also be further noted that for pre-insulated rigid steel pipes, the medium pipe, the insulation material and the jacket tube form a tightly connected unit, that is, the pipes expand as a whole unit. As a result, the earth pressure (covering), that weighs on the pipes hinders the expansion. The obstruction of the expansion leads to a compressive stress on the steel pipes, which must not exceed the permissible stress of 190 N/mm². In order to ensure these conditions, cold laying methods are conducted, such as cold laying method 1 and 2 or the laying with thermal pre-stress (overview in Table 7.6).

Table 7.6 Overview, advantages, and disadvantages of the laying of earth-laid pre-insulated rigid steel pipes.

Laying		Advantage	Disadvantage
Cold laying Method 1	Cold laying	low axial stresses due to thermal expansion Pipe trench can be filled immediately	maximum allowable operating temperature ≤ 85 °C
	Conventional	maximum permissible axial stress is not exceeded Pipe trench can be filled again immediately	Permissible installation length must be maintained by arranging the necessary extension limb (L, Z, and U).
Cold laying Method 2	Relates to: Self prestressing	Pipe trench can be filled immediately Saving of expansion legs possibly also possible in the sliding area	Extremely large stretching movements danger of buckling Axial stresses exceed the expansion limit of the material Later on integrated start drill branches not possible
Thermal Prestress		Limitation of the axial prestress Various laying length Low axial expansion Saving of expansion legs	The pipe trench must be kept open until the prestress has been completed. Depending on the method, an adjustable operating medium or a 380 V

7.5.3.1 Cold laying method 1

This method does not exceed the maximum installation length specified by the manufacturers. When this length is reached, an expansion bend is installed to accommodate the expansion. Approximately in the middle between two arcs, a natural anchor point (NAP) is created, from which the pipelines expand in both directions (Figure 7.19). The maximum installation length depends on the design temperature, the pipe dimension and the earth covering.

Note: At a maximum operating temperature of 85 °C, the permissible stress of 190 N/mm² is never reached. As a result, there is no restriction on the length of the installation. However, in the static design the temperature must not be higher than 85 °C.

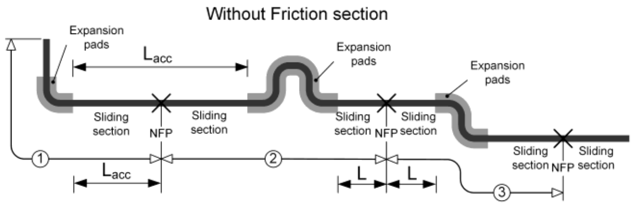


Figure 7.19 Example: Route with natural anchor points and expansion pads at compensation elements based on “AGFW-Merkblatt FW 401 Teil 10” [100].

The cold-laying method 1 has the disadvantage that relatively many fittings have to be installed in the form of elbows, which makes laying more expensive. On the other hand, miter seams (3° bends) and the subsequent installation of T-pieces as well as boreholes have no influence on the statics.

Application: for example, areas with many changes of gradient - or direction, site development pipelines (branch lines), house connections.

7.5.3.2 Cold laying method 2

This is a cold laying method without taking into account maximum installation lengths specified by the manufacturers. The process is also referred to as 'operational self-pre-stress'. With the commissioning, the permissible yield limit of 235 N/mm² is exceeded and the pipes are plastically compressed. On the one hand, this means that stronger extension limbs and expansion pads are required. On the other hand, the T-outlets must be reinforced and welding seams (3° bends) as well as the subsequent installation of T-pieces as well as spot drills are not permitted.

The method involves a more cost-effective installation compared to the cold laying method 1.

Application: e.g. transport pipelines outside residential areas, areas with "permanent” structures (e.g. large exiting structures overhead, housing cooperatives, house connections, etc.).

7.5.3.3 Thermal pre-stress

If the pipelines are subject to thermal pre-stress, then there is no restriction of the laying length. The permissible stress is not exceeded. As a rule, the finished pipeline is heated to the pre-stress temperature T_{Pre}. The pre-stress temperature T_{Pre} corresponds to the laying temperature T_{Lay} subtracted by half of the pipe-static design temperature ΔT_{PSD} and is calculated as follows:

$$T_{Pre} = T_{Lay} + \frac{\Delta T_{PSD}}{2} = T_{Lay} + \frac{T_{Des} - T_{Lay}}{2} = \frac{T_{Des} + T_{Lay}}{2}$$

As soon as the pipe is heated and has thus expanded, the pipeline is sanded and the trench is filled. The pipeline can be cooled again after trench filling. The sliding range of the pipes is relatively short and consequently also the extension limb and expansion pad lengths (Figure 7.20).

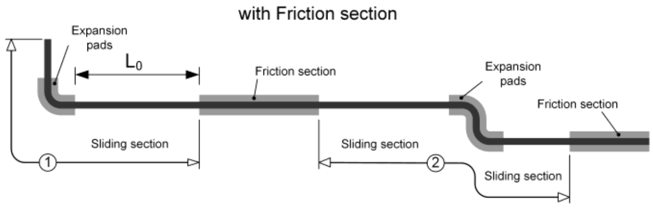


Figure 7.20 Example route with slide and adhesion zone and expansion pads with compensation areas based on the “AGFW-Merkblatt FW 401 Teil 10” [100].

The thermal pre-stress requires long construction stages (> 100 m) and thus affects the construction time. Compared to cold-laying, however, the pipeline construction is more cost-effective. Miter seams (3° bends) and subsequent T-pieces and start drills are statically unproblematic.

Application: e.g. transport pipelines, residential areas with few changes in slope or direction, development pipelines.

7.5.3.4 Double and multiple pipes

In the case of pre-fabricated and pre-insulated multiple pipes or double pipes, the pipes are tightly bonded at elbow ends and end points. Statically non-significant spacing blocks are inserted between the pipes. For pipe structural reasons, this means that the maximum temperature difference between the individual pipes is limited to a maximum of 100 K. This maximum temperature difference must be taken into account, especially when starting up from cooled-off systems.

7.5.4 Exposed piping

Exposed piping refers to freely laid pipes in basements, car parks, power line tunnels, but also earth-laid pipes in concrete ducts. Steel jacket pipes described in chapter 4.3.1.5 are statically seen as freely laid as well.

In contrast to the earth-laid systems, freely laid pipes are only marginally impeded by friction losses at pipe bearings. In theory, the extension of the ΔL can be as large as necessary if the corresponding measures for strain absorption are followed. However, these measures become increasingly difficult to carry out at dimensions from

DN 100 and at high temperatures. It should also be noted that the pipes can move in all directions. The following are ways to absorb the strains.

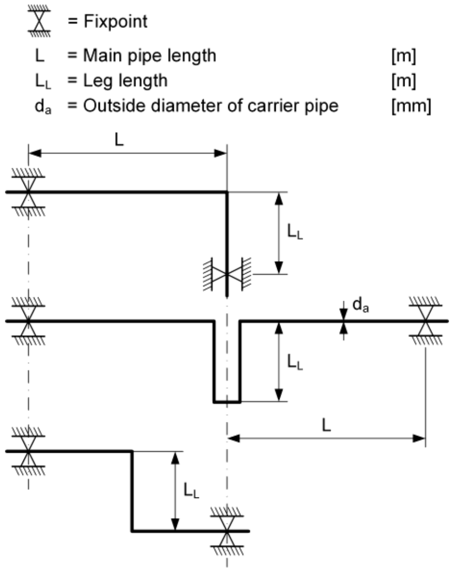


Figure 7.21 Typical shapes of extension limbs of exposed pipes (Representation refers to [49])

To minimise length expansion, natural expansion compensation is made possible by changing direction of the pipelines. In the case of freely laid pipelines, expansion limbs are also realised as angular arcs (L-arc), double angular arc (Z-arc) and U-arc (Figure 7.21).

According to [49] the extension of the projection length L_L is not only dependent on the expansion ΔL or the main pipe length L, but also increases with increasing pipe dimension or the static pipe design temperature PSD. This means that the larger the expansion ΔL and pipe dimension is, the larger the required extension limbs and the longer the outlet length L_L.

For exposed laid pipes, the following equation according to [49] can be used for the rough calculation of the discharge length L_L:

$$L_L \approx \frac{\Delta T_{PSD}}{64} \sqrt{L \ d_a}$$

The diagram in Figure 7.22 gives an indication for the design of the expansion compensators for the above equation for determining the projection length L_L or open, building or duct-laid systems normalized to a maximum temperature difference of 100 °C. The expansion compensators are designed for the following purposes. The projection length L_L of the extension limbs can be converted proportionally for other temperatures on the basis of the diagram in Figure 7.22.

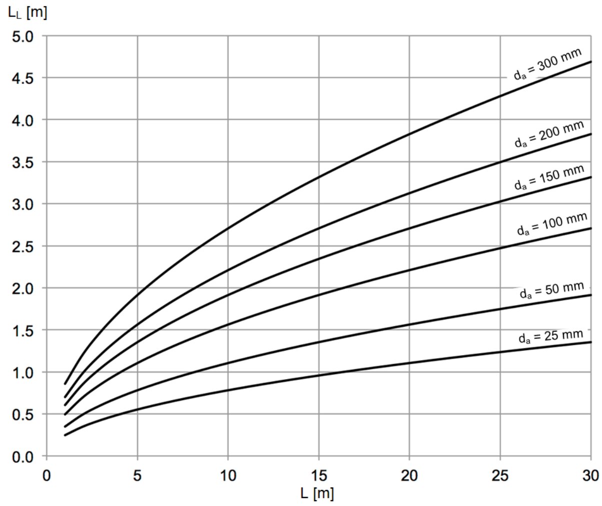


Figure 7.22 Expansion leg lengths for hot-rolled ferritic and austenitic piping steels for free-laying, building or duct-laid pipe systems standardized to a maximum temperature difference of 100 °C (calculated according to [49]).

Alternatives to natural expansion compensation for freely installed systems are lateral, axial and joint compensators, which can absorb targeted deformation by means of flexible bellows. The “AGFW Merkblatt FW411” [102], provides design principles for systems installed in open spaces, buildings, and ducts. Included are tables and diagrams for static design up to a diameter of DN 80. For larger nominal widths, proof must be provided for the stress caused by thermal expansion, the dead weight and the load on the supports, as well as proof for anchor points, bearings, bends, T-junctions, bores, extension limbs, spans. The basic principles for the static calculation are provided by SN EN 13480 [93], EN13941 [94] and AD-2000 [119].

7.5.4.1 Pipe mount

Pipes are provided with pipe clamps for support. They are connected to ceilings, walls or floors and anchored. The larger the pipe dimension, the fewer supports are needed. Basically, there are four different types of pipe brackets or pipe bearings, which are used in freely laid systems.

- **Anchor Points** (Fixed points): No movement in one direction is possible for the pipes. The anchor points can assume massive dimensions for large sizes. Special designs are often required.
- **Slide Bearing**: The pipes are able to move longitudinally. In addition, there is a small margin for lateral and up/down displacements. This type of bearing is by far the most frequently used.
- **Lateral Expansion Bearing** (Cross-lateral expansion bearing): The pipes are able to move in a longitudinal and transversal direction. In addition, there is a small clearance for displacements and up/down movement. This length type is used in the area of expansion legs when the expansion has to be absorbed from two sides.

- **Guide Bearings:** These bearings are special slide bearings which allow movement only in the longitudinal direction. Guide bearings are only required when using compensators.

The "AGFW-Merkblatt FW 411 [102] describes a selection and arrangement of pipe supports for steel medium pipes with thermal insulation made of fibre insulating material. The suitable positioning of slide bearings in the area of the expansion compensators is also listed. The "AGFW-Merkblatt FW 410" [101] applies to earth-laid and exposed steel jacket pipes.

Due to reasons of process engineering, low deflection limits are set when determining the pipeline supports and the pipeline stresses resulting from the dead weight. For the strength assessment, a simplified verification procedure referred to as span width control, can be used. Because of their own weight, it is important whether the pipes are filled with water or gas or whether they are insulated. For steel pipes with a diameter of 50 to 500 mm, a density of the flow medium of 1000 kg/m³ and a constant wall thickness ratio of di/s ≈ 30, the following relationship results for the span widths:

$$L \approx f\ d_i^{\frac{2}{3}}$$

f = 0.3 for an empty and uninsulated pipe

f = 0.23 for a filled uninsulated pipe

f = 0.2 for a filled and little isolated pipe

The equation is shown in Figure 7.23. From the DN 500, the span for limiting the stress at the support point remains constant.

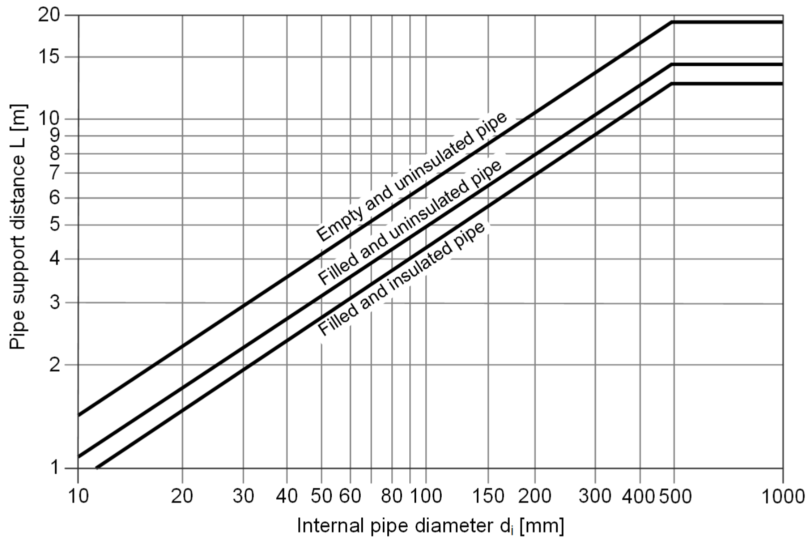


Figure 7.23 Guideline values for the maximum permissible pipe span L as a function of the pipe routing form and the inner diameter of the steel pipe di (pipes in the building) according to [49].

7.5.4.2 Laying with pre-stress

For freely laid pipes (inside as well as outside), it is recommended to pre-stress pipes from DN 100. A pre-stress between DN 50 and DN 80 can be beneficial. Even smaller dimensions are usually so elastic that the strains can be easily absorbed. The pre-stress of freely laid pipes is invariably a cold pre-stress, i.e. the pipe lengths between the elongation ranges, are shortened by the preload dimension (Figure 7.24). The preload dimension usually corresponds to half the length elongation at a temperature difference between the laying temperature T_{Lay} and the design temperature T_{Des}.

With a cold pre-stress, the designed length of extension limbs can be reduced. The connection of the two extension limbs or the welding is carried out by a mechanical preload. In this case, an extension limb is pressed to the shortened limb until the gap is compensated and the two ends can be welded together.

If these measures are not sufficient to accommodate the strains due to limited spatial conditions, compensators can be used to remedy the situation, and these systems can also be pre-stressed.

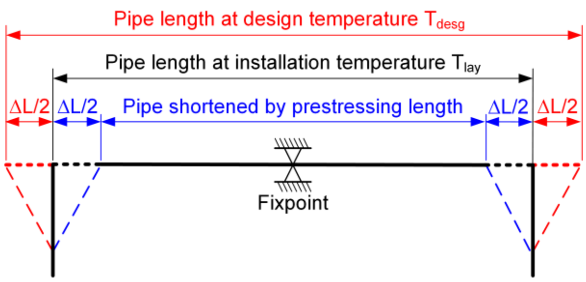


Figure 7.24 Schematic representation of the cold pre-stress of freely laid pipes in the range of the extension limb (simplified representation of only one extension limb).

Note: Laying methods described above are common for high temperature water or steam systems. A detailed description with compensator systems is not included in this handbook. References based on the "AGFW Arbeitsblatt FW 401 and [100] and FW 411 [102] and to special literature are made.

7.5.5 Flexible pre-insulated PEX pipes

As flexible pre-insulated PEX pipes have a much higher elasticity than steel pipes, they are self-compensating, i.e. the elongation is absorbed by the pipes themselves by increasing the diameter. Although the expansion coefficient PE is about 20 times higher than that of steel, expansion must be prevented. In the case of underground pipes, this is possible without any problems due to the pressure of surrounding earth. Expansion pads can be completely neglected. In the case of freely laid flexible pre-insulated PEX pipes, longitudinal elongation is prevented by anchor points before arcs (Figure 7.25). The bearings between the anchor points are only used for fixing the pipes. Plain bearings and compensators are not used in flexible pre-insulated PEX pipes.

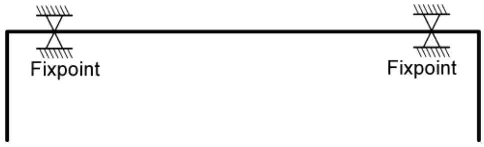


Figure 7.25 Arrangement of the anchor points of flexible pre-insulated PEX pipes before arches and by changes in direction.

8 Heat Transfer – Technology and Hydraulic System

This chapter describes the components and technical basics of a house substation. In addition, the hydraulic system for heat transfer is discussed and the most important standard circuits for district heating networks are described.

8.1 Components and Technology of a House Substation

This chapter elaborates on the explanations made in chapter 5.2 and relates to Figure 5.2 and Figure 5.3. Depending on the connection load and design standard, the house substation can be installed as a unit (compact station), which contains the components of the transfer station and the consumer installation, or the consumer installation and the house substation can form a separate unit.

A compact station must include the following components:

- Connecting and terminating fittings within the supply- and return (at the primary and secondary site)
- Draining or ventilation possibility (depending on the gradient of the connecting pipes)
- Heat measuring device (volume flow measuring device, temperature measurement, calculator)
- Indicator of temperature and pressure measurement
- Differential pressure and volumetric flow rate control (pressure independent control valve)
- Heat exchanger
- Pump/s
- Dirt traps

The following additional components are possible depending on the plant size:

- Short-circuit connection
- Counter bypass
- Additional display of temperature and pressure
- Additional differential pressure and volumetric flow rate controls (pressure independent control valves).
- Additional heat exchanger

The heat supplier adheres to the standard in the technical connection regulations (TCR). The combination options with the consumer’s installation must also be coordinated with the heat supplier, which is essential for a compact station. Certain functions are often used by both parties. For example, control valves with emergency control function can secure the customer's system.

8.1.1 Heat Load Demand

The maximum heat output of the customer agreed in the heat supply contract is usually set and sealed at the throttle of the volumetric flow rate limiter during commissioning. This limits the maximum volumetric flow rate. The heat supplier should, if necessary, check the purchased capacity on a random basis.

8.1.2 Materials and joints

The selection of materials for the primary-sided components shall be carried out in accordance with DIN 4747-1 [115], whereby, contrary to DIN 4747-1, non-ferrous metals and soldered heat exchangers with non-ferrous metal solder are often not permitted. The fasteners and seals used shall be suitable for the operating conditions in terms of pressure, temperature and heat carrier medium.

The pipes and fittings shall be coated with a temperature-resistant corrosion protection. Automatic vents, pressed connections, rubber compensators, conical connections and hemp as sealing material are not provided.

Further requirements and deviations from DIN 4747-1 must be listed in the Technical Connection Regulations.

8.1.3 Thermal Insulation

Thermal insulation must be resistant to aging and does not have a corrosive effect on the parts of the installation when wet. It must also be chemically and dimensionally stable at operating temperature. The installation and disassembly of the heat meter and the associated sensors must be possible without incursion of the thermal insulation. Supply and return pipes are generally separate and insulated appropriately. The thermal expansion of the pipes must not affect the thermal insulation. Thermal insulation is to be installed with full butt joints and staggered joints. The longitudinal and impact joints must be completely closed with a suitable sealant.

The primary-side pipes must be insulated e.g. with glass fibre shells (raw density at least 80 kg/m^3, thermal conductivity λ = 0.034 W/(m K) at tm 50 °C and 0.039 W/(m K) at tm 100 °C): The laid pipes must be protected additionally with a corrosion- and temperature-resistant sheathing.

The minimal thermal insulation classes for heating pipes installed in buildings, in unheated rooms and for hot water pipes are presented in Table 8.1. The presented thermal insulation classes are valid for an operating temperature up to 90 °C. At higher operating temperatures, the thermal insulation class must be increased appropriately and at maximum supply temperatures of 30 °C the thermal insulation classes can be reduced.

Table 8.1 Minimum thermal insulation classes for heating pipes laid in buildings in unheated rooms and hot water systems.

Nominal Diameter DN	Thermal Insulation Class [mm] for $0.03 > \lambda \leq 0.05$ W/(mK)	Thermal Insulation Class [mm] for $\lambda \leq 0.03$ W/(mK)
10-15	40	30
20-32	50	40
40-50	60	50
65-80	80	60
100-150	100	80
175-200	120	80

8.1.4 Heat Meters

The use of calibrated heat meters is necessary for the invoicing of fuel deliveries or the heat obtained from a customer. The heat measurement requires a flow and a temperature difference measurement between the supply and return.

The accuracy class of a heat meter is determined by the measurement accuracy of flow rate and temperature difference.

The measuring range of the flow rate is given by the working range between nominal flow q_p and minimum flow q_i. The ratio of nominal flow to minimum flow is the bandwidth of the flow range in which a certain accuracy of the volumetric flow rate measurement is guaranteed. Water quality also has a major impact on measurement accuracy during long-term use.

The pressure drop at nominal flow rate must be taken into account.

The following measurement methods are used for flow measurement (see also Table 8.2):

- Magnetic-inductive flow rate measurement
- Ultrasonic flow rate measurement
- Static flow rate measurement based on the oscillating jet principle
- Mechanical flow measurement by means of impeller or turbine wheel.

8.1.4.1 Implementation of heat meters

To achieve the required measuring accuracy, the following instructions must be observed [21]:

- Observe the installation instructions of the heat meter supplier (inlet and outlet section, horizontal/vertical installation, sensor installation, etc.)
- The inlet and outlet distances vary depending on the nominal size and technology. As a guide value, the inlet pipe has an inlet section of 5 x DN and the outlet section 3 x DN.
- The temperature sensor for the return flow must be installed downstream of the flow meter.
- If possible, the flow meter should be placed between two shut-off devices. This facilitates maintenance work and meter replacement according to the calibration cycle.
- Design for a temperature difference > 20 K. Temperature difference during operation at least 3 K.
- Uniform temperature distribution over the pipe cross-section upstream of the temperature sensors (if necessary, install a static mixer).
- Stable control circuits (oscillating controllers can cause large measuring errors).
- Disturbances due to incorrect circulation are minimised if the temperature difference measurement is on the same level as the flow measurement (incorrect circulation is at least measured correctly).
- Operate the heat meter only in the permissible flow rate range qp to qi.
- The flow rate must correspond to the minimum flow rate of the heat meter at minimum opening of the control valve.
- Compact heat meters are advantageous because interference with the short signal transmission from the sensor to the transmitter and the calculator is practically eliminated.
- The heat meters are commissioned in a technically perfect manner and, if necessary, interference sources are systematically identified by specialists.
- It is not permitted to extend the sensor cables. Heat meters are calibrated and calibrated including sensors.

For magnetic-inductive flow metering devices:

- There must be no magnetite in the water, as this is deposited on the measuring probes and thus influences the measurement (reduction of the flow measurement value).
- In existing plants, the required water quality (clear water) can be achieved with a magnet sludge separator in conjunction with a degassing device.
- In the case of new plants, care must be taken to ensure that the water is sufficiently degassed from the outset and that the oxygen content is reduced to zero.
- If large measuring errors occur, the inner walls of the flow meters must be cleaned, which does not mean the cause (e.g. foul water) has been eliminated.
- Since a very low voltage is applied via the measuring probes (a few millivolts), the measuring method is sensitive to electrical interference fields, especially with split devices (separated sensor and measuring transducer). This can be mitigated by using compact flow sensors.
- Only use shielded and twisted pipes and avoid the proximity of strong magnetic fields from electric motors or frequency converters.

For ultrasonic flow meter devices:

- Contamination of the deflecting mirrors with small nominal diameter and gas inclusions in the water can cause measuring inaccuracies.
- The avoidance of these disturbing influences requires a high water quality as well as sufficient degassing of the water, which prevents deposits on the deflecting mirrors.

For oscillating jet flowmeters:

- Basically, insensitive to contamination, since only a partial flow with increased flow velocity is necessary for the measurement.
- For horizontal installation, make sure that the measuring head is mounted laterally (not at the top or bottom). For vertical installation, no special measures have to be taken.

For mechanical flow metering devices:

- Installation of a dirt filter upstream of the water inlet of the flow meter to prevent damage to or clogging of the impeller-/ turbine wheel.
- Regular revision to rule out wear and tear as a source of error.
- Careful design to ensure that the operating flow rate does not fall below the minimum flow rate Qmin when the volumetric flow rate is variable or that it only falls below the minimum flow rate in exceptional cases.

8.1.4.2 Maintaining the measuring stability

The heat meter must be installed and operated in accordance with the Measuring Equipment Regulation ("Messmittelverordnung MessMV" [64]) and the "Verordnung des EJPD über Messgeräte zur thermischen Energie" [65]. The heat meter is usually monitored and maintained by the heat supplier. The measuring stability of the heat meter must be ensured by the heat supplier.

In accordance with the Measuring Equipment Regulation, a calibration must be carried out every five years by an authorized body. However, if at least 150 heat meters are in operation during a thermal composite and this measurement data is monitored during operation, an application may be submitted to METAS (Swiss Federal Institute of Metrology) for a procedure by which the calibration period can be extended under the following conditions:

The procedure must be suitable to ensure correct measurements.

- All heat meters used must be operated in accordance with [65] and none of these meters may be used for more than ten years at any time without recalibration.
- Defective meters must be replaced by compliant meters.
- All meters used must be operated under comparable operating conditions.
- The user informs METAS about the results of the procedure once a year.

If the nominal power of the heat measuring device is at least 10 MW, a calibration can be carried out in accordance with [65] to maintain the measurement stability. However, the following conditions must be met:

- The thermal energy is determined between two permanent partners via fixed supply lines in a measuring station from the measurement data of one or more heat meters, whereby the sum of the nominal outputs of the heat meters used is at least 10 MW.
- The heat measuring device is used between two trade partners who are basically able to assess the measurement results.
- The heat measuring equipment is subject to regular measurement supervision by the expert operating personnel.
- If parts of the measuring system cannot be calibrated on site, the measuring instruments shall be calibrated by a calibration laboratory recognized by METAS, an authorized calibration center or by METAS. After calibration, the heat measuring equipment is secured with identifiable operating seals.
- The calibration of the heat measuring device must be carried out as required, but as a rule every 12 months. All parts of the measuring equipment shall be calibrated at intervals of maximum two years.
- A record is kept of the work carried out on the heat measuring device (maintenance, adjustment, calibration). The entries must show which work was carried out, when and by whom. In the event of complaints, it must be possible to submit the records to the competent body.

Table 8.2 Assessment of main the flow measurement methods [21].

	Magnetic-inductive	Ultrasonic	Oscillating jet	Mechanical
Ratio q_p/q_i	100-150	100-150	25-100	25-100 60-190*
Pressure loss by nominal discharge q_p [kPa]	7-15	7-20	9-25	10-15
Measuring Accuracy	High	High	High	Medium
Sensitivity of measurement accuracy to water quality	High	low to moderate**	Low	Low
Wear and tear / Service expense	Low	Low	Low	High
Sensitivity of measurement accuracy to electrical interference fields	High	Low	Low	low to moderate***

* Special version (Woltmann meter)
** Contamination of the deflecting mirrors with small nominal diameters
*** With inductive pulse generator

8.1.5 Pressure safety

The pressure safety for the direct connection can be guaranteed without pressure reduction if the maximum pressure in the network p_{N_max} is less than or equal to the permissible pressure of the house substation or the domestic heat distribution system p_{HD_acc}. Otherwise, pressure reduction and pressure safety measures are required, as demonstrated in Figure 8.1 and

Table 8.3 according to DIN 4747-1 [115].

For the indirect connection, the primary side of the heat exchanger must be measured for the maximum network pressure p_{N_max}. The arrangements of the safety devices against exceeding the secondary-sided operating overpressure shall be made in accordance with SN EN 12828 [98] or DIN EN 12953 [108]. With regard to the pressure vessels to be secured, "AD 2000 vom Merkblatt A2" [119] must be observed. For this purpose, each heat exchanger must be protected on the secondary side by safety valves against exceeding the permissible operating pressure. The following points must be taken into account:

- Connection of a maximum of three safety valves per heat exchanger.
- Vertical installation with rise as short as possible, max. 1 meter.
- Pipes installation without shut-off, dirt traps and fittings that lead to a narrowing of the cross-section.

- Arrangement in an easily accessible location.
- Vertical installation only.
- Information sign.
- Separate discharge pipe on a slope, pipe opening accessible.
- Safety valve must be component tested.

If it is possible to heat up the heat exchanger through the secondary side, pressure protection measures must also be taken on the primary side. Each heat exchanger must be connected to at least one expansion vessel.

The design and mass of the membrane safety valves for the secondary side must be in accordance with the information in

Table 8.4.

8.1.6 Temperature safety

Temperature safety at the domestic installation depends on the maximum permissible house system temperature and the maximum primary operating medium temperature. In DIN 4747-1, requirements according to Table 8.5 are set.

If a hot water heating system is connected downstream of a substation or a system for room heating with supply temperature control and temperature protection, the primary operating medium temperature and not the maximum primary supply temperature is decisive for the design of the safety equipment for temperature protection of the hot water heating (Table 8.6).

In Table 8.5 and Table 8.6 control devices with safety functions are described. The control, monitor and limit impulse can act on a common actuator if it is classified. As type-tested actuators, volumetric flow rate meters without external energy, which are equipped with an actuator with safety function, can also be used.

The setting range of the safety equipment shall not exceed the maximum permissible temperature of the domestic installation by more than 10 %, limited to a maximum of 5 K.

The control unit shall be designed in such a way as to avoid the risk of evaporation or emptying at the consumer's installation or the domestic heat distribution system. In the case of hot water heating systems, the control valve should be installed in the supply line.

In order to measure the temperature as quickly and efficiently as possible, the temperature sensors for the control, monitor and limit impulse should be installed in such a way that the mixing temperature can be reliably determined. In the case of indirect connection, the temperature sensors should be arranged in or as close as possible to the heat exchanger.

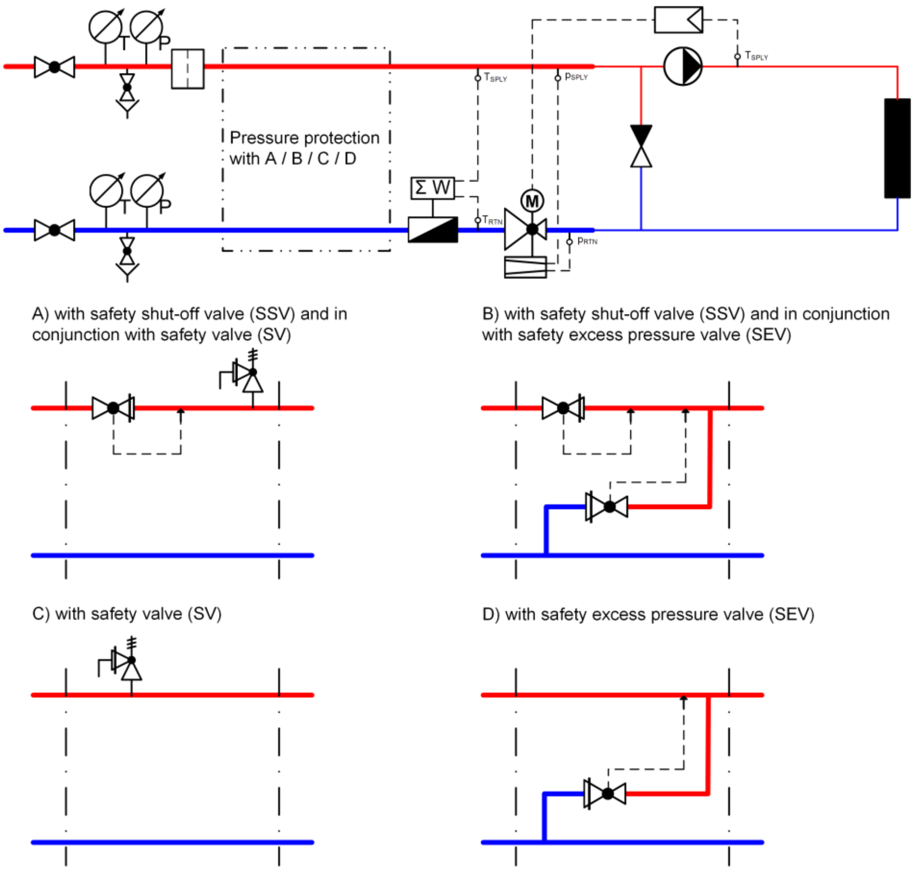


Figure 8.1 Principle diagram for equipping a house substation with direct connection with safety devices for pressure protection with pressure reduction based on (DIN 4747-1 [115]). See also Table 8.3.

Table 8.3 Safety equipment for pressure protection based on DIN 4747-1[115].

Permitted operating pressure of the domestic heat distribution	Direct Connection					Indirect Connection (secondary)	
	SSV with SV*	or SSV with SOV*	or SV	or SOV	Expansion vessel	SV	Expansion vessel
≥ p_{Nmax}	not required	not required	not required	not required	not required	required	required
< p_{Nmax}	required	required	required	required	not required	required	required

SSV = Safety shut-off valve, component tested

SV = Safety valve, component tested

SOV = Safety overflow valve, component tested

* SV- and SOV-Design for 1 % in minimum of the k_{VS}-value of the safety shut-off valve

Table 8.4 Selection of diaphragm safety valves against excess pressure due to water expansion in indirect connections according to DIN 4747-1 [115]. This applies to a response pressure of ≤ 3.0 bar.

		Discharge capacity of water in l/h = nominal heat output in kW		
		≤ 100	≤ 350	≤ 900
Supply Pipe	Minimum diameter respectively Minimum nominal diameter DN	15	20	25
	Connection thread*	G 1/2	G 3/4	G 1
Discharge Pipe	Minimum diameter respectively Minimum nominal diameter DN	20	25	32
	Connection thread*	G 3/4	G 1	G 1 3/4

* Pipe thread relates to DIN ISO 228-1

Table 8.5 Safety equipment for temperature protection of house substations for space heating according to DIN 4747-1 [115].

Network Operation	Maximum primary operating temperature	Maximum admissible temperature domestic Installation	Operating medum temperature Control	Temperature control TC[a]	Safety Temperature Monitor STM[a]	Safety Operation based on SN EN 14597 [97]
steady network operation	≤ 120 °C	≥ primary supply temperature	required	not required	not required	not required
		< primary supply temperature	required	not required	required	required
	> 120 °C	< primary supply temperature	required	required	required	required
Variable and variable constant network operation	≤ 120 °C	≥ primary supply temperature	not required	not required	not required	not required
		< primary supply temperature	required	not required	required	required [c, d]
	> 120 °C ≤ 140 °C	< primary supply temperature	required	not required	required	required [c, d]
	> 140 °C	< primary supply temperature	required	required	required	not required

a) Definition based on SN EN 14597

b) Decentralized temperature control with thermostatic radiator valves

c) Not required for systems whose primary heating water volumetric flow rate does not exceed 1 m³/h. If the STM is no longer required, a TC will be needed. Panel heating systems and domestic hot water heating systems are excluded from the relief.

d) Based on SN EN 14597, the control valve meets the requirement for internal tightness (0.05 % of k_{VS} value).

Table 8.6 Safety equipment for temperature protection of house stations for domestic hot water heating according to DIN 4747-1 [115].

Network Operation	Maximum primary operating temperature	Maximum admissible temperature domestic Installation	Operating medum temperature Control	Temperature control TC[a]	Safety Temperature Monitor STM[a]	Safety Operation based on SN EN 14597 [97]
constant, variable and variable-constant Network Operation	≤ 100 °C	≤ 75 °C	required	required	required max. T_{Hzul}	required
		> 75 °C	required	not required	required	not required
	> 100 °C ≤ 120 °C	≤ 75 °C	required	not required	required max. T_{Hzul}	required
		> 75 °C	required	required	not required	not required
	> 120 °C	≤ 75 °C	required	required	required max. T_{Hzul}	required [c]
		> 75 °C	required	required	required max. 75 °C[b]	required [b c]

a) Definition based on SN EN 14597

b) Not required for hot water heating systems with flow-through water heaters whose primary heating water volumetric flow rate does not exceed 2 m^3/h.

c) Based on DIN 32730, the control valve meets the requirement for internal leak tightness (0.05 % of the k_{VS} -value).

d) The hot water temperature can already be controlled by the safety equipment.

e) If the safety function according to SN EN 14597 is required, a control valve (primary) that already exists for space heating can be used.

8.1.7 Control Unit

A pressure independent control valve with safety function must be used to control the secondary-side supply temperature (through-way control device with integrated differential pressure regulator/volumetric flow rate limiter). The differential pressure regulator ensures a constant pressure difference above the control organ, which results in a high valve authority. The volumetric flow rate limiter is used to set the system to the service agreed in the heat supply contract. The differential pressure of the pressure independent control valve is 20 kPa. From a control point of view, the required minimum heat meter flow rate must be guaranteed (valve opening). When the power is disconnected, the safety function closes the valve, avoiding pressure surges.

The secondary-side operating medium temperature of the consumer's installation can be regulated, for example, externally. The primary return temperature shall be limited to the maximum permissible return temperature based on the heat supply contract.

If the return temperature is too high, two functions are allowed. Either the primary-sided pressure independent control valve is closed or the primary-side return temperature is used as the control value instead of the secondary-side operating medium temperature while the return temperature is too high.

The sensors for measuring the secondary-side operating medium temperature and the primary-side return temperature must be placed immediately next to the outlet of the heat exchanger. Contact sensors should not be used for this.

8.1.8 Maximum return temperature

The return temperatures specified in the technical connection regulations are to be understood as maximum values; if possible lower return temperatures should be sought.

The maximum return temperature should never be exceeded. Exceptions may be granted for hot water charging, but should be limited to a short period of time. The conditions for the maximum return temperatures must be recorded in the heat supply contract (Technical Connection Requirements TCR).

Appropriate measures to reduce the return temperature include generous heat exchanger surfaces, flow velocity or storage charging systems for domestic hot water heating, radiator thermostat valves, limitation of the hot water circulation quantity, as well as a hydraulic system appropriate for the consumer's installation and the house distribution system.

8.1.9 Heat Exchanger

The regulations for the safety of pressure equipment (Pressure Equipment Regulation) and declaration of conformity [66] must be observed for the strength-based design. The execution of the heat transfer should in principle be carried out according to the counter-current principle.

The design pressure, the design temperature and the least temperature difference (LTD) shall be specified in the technical connection requirements. A least temperature difference of 4 to 5 K under design conditions (lowest outside temperature) and a least temperature difference of ≤ 3 K in the partial load range are to be defined as target value. As a result, the heat exchanger has a sufficiently large transfer surface.

The following types of heat exchangers are possible:

- Plate heat exchanger (screwed, soldered or welded)
- Shell and tube exchanger
- Spiral-plate heat exchanger

If soldered heat exchangers are not permitted because they are made of copper or nickel or for other reasons, this must necessarily be specified in the technical connection requirements.

The heat exchangers must be installed mechanically without tension. No axial forces and bending torques may occur in the connections to the heat exchanger.

The secondary-sided heat exchanger outlet temperature should not be too high and not too low. Recommended is an outlet temperature plus around 5 °C to the highest required supply temperature of all connected heating and hot water circuits.

8.1.10 Grounding

The transfer station and the consumer's installation must be connected to the potential compensation. In particular, low-voltage elements such as the bus system should also be equipped with surge protection.

8.1.11 Data Collection

If possible, a direct or indirect data connection to the heat station (bus connections such as modbus, etc.) must be integrated into the house substation.

This essentially simplifies invoicing, fault recording and plant optimisation.

8.1.12 Dirt Trap

A dirt trap covering a large area with support and fine filter and the following properties should be installed in the primary-side supply line:

- Support filter clearance mesh size 0.8 mm
- Fine filter clearance mesh size 0.25 mm

8.2 Domestic hot water generation

The domestic hot water heating of drinking water is possible all year round, provided that the heat supplier operates the district heating network throughout the year at the temperature necessary.

According to the supply temperature, the pressure ratio in the primary network and the toxicity of the circulating water (fluid category 3 or 4 according to chapter 4.8.1), an adequate heat transfer between circulating water and domestic hot water must be taken into account.

If the circulating water corresponds to fluid category 3, the separation between circulating water and hot water can be done in a single-walled manner.
However, category 4 fluids must be separated in a double-walled manner. A double-walled separation consists of at least two solid and sealed areas or containers, which range from a neutral intermediate zone between the hot water on one side and the circulating water on the other. The intermediate zone may contain a gas, inert porous material, or a fluid of category 1, 2 or 3 and may be connected to a visual or audible alarm [88].

In Switzerland the SIA 385/1 [83] and SIA 385/2 [84] standards shall be applied to the implementation of domestic hot water supply systems with drinking water in buildings. In the case of heat supply with district heating, additional attention must be paid to the following points:

- Specifications of the technical connection regulations
- Minimum and maximum primary operating medium temperature
- Maximum permissible primary return temperature.

The system of domestic hot water heating shall be designed to the minimum primary-sided operating medium temperature (transition time and summer) and to the maximum primary-sided operating medium temperature. This applies in particular to the dimensioning of the heat exchanger and the control system. The entire system must be further designed to the lowest possible primary return temperature. There are basically three different variants for domestic hot water heating:

- Instantaneous water heater
- Hot water storage with internal heat exchanger
- Hot water storage with external heat exchanger

Table 8.7 Methods of domestic hot water generation.

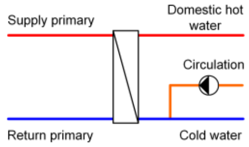
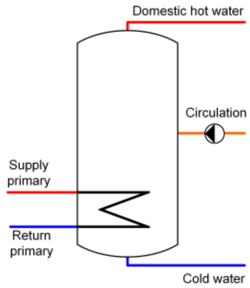
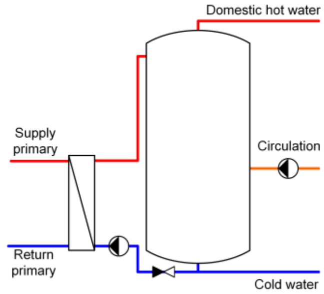
	Instantaneous water heater	Hot water storage with internal heat exchanger	Hot water storage with external heat exchanger
			
Description	With the di instantaneous water heater, the hot water is heated directly at the time of need via a plate heat exchanger. Connection usually on the primary side. An upstream heat storage is recommended to reduce the load peaks (not shown). Additional possibility of return flow switchover to the middle half of the heat storage tank (not shown), to improve the return flow temperature in circulation heating Provide a switch-off sensor.	Water heater in the form of a tank with built-in heating surfaces in which the cold water is heated and stored. Peak loads are covered by the storage tank. Connection usually on the secondary side. Provide a switch-off sensor Variable loading volume for volumes greater than 500 litres Pay attention to large register with loop in the bog. Circulation can also be integrated via a small external heat exchanger.	This variant represents a combination of direct flow and storage principle. The storage tank is charged via an external heat exchanger and loading pump (circuit with hot water). Peak loads are covered by the storage tank Provide a switch-off sensor.
Asset	Low return temperature Low cost Low space requirement standby losses (without heat storage) Reduced legionella problem	High draw-off quantity possible Insensitive to lime scale Low control requirements	Low return temperature High tapping quantity possible Low constant charging power (reduction of connected load) High degree of storage utilization
Drawback	High connection load required (negligible with upstream thermal energy storage) Heat losses with a upstream thermal energy storage needs a good (complex) control sensitive to high lime content in water	Rising return temperature during the charging process Heat losses of the storage tank Decreasing heating capacity during the charging process legionella problem	High investment costs requires a complex control system Heat losses of the storage tank legionella problem
Possible return temperature	< 30 °C	45 °C	< 35 °C

Table 8.7 describes variants that can be connected directly to the district heating network (primary side) or indirectly (secondary side) via a heat exchanger.

With the instantaneous water heater, for example, the relatively high connection load can be compensated by an upstream hot water storage tank and the connection is also secondary. Otherwise, the connection is only useful from an exergy point of view if it is done on the primary side. However, the disadvantage is that a high connection load is required.

The storage systems are also suitable for connection on the secondary side (after the main heat exchanger). The storage systems, in all the storage water heaters with internal heat exchanger, should be designed as a hot water priority circuit if possible. The priority circuit allows a lower subscribed connection load to be defined. By means of an intelligent control system, the power in the heating circuit is reduced only to the extent that the subscribed connection load according to the heat supply contract is not exceeded and sufficient power for domestic water heating is provided. If possible, hot water heating should also occur in the night hours.

In the case of hot water storage with external heat exchanger, consistently low primary return temperatures are possible. During heating and during hot water and circulation water withdrawal, care must be taken to ensure that the cold and hot water is layered properly.

With an additional heat exchanger for preheating the cold water, the primary return temperature can be further lowered and is suitable as a possibility for all three above-mentioned variants of hot water heating.

8.3 Legionella Problem

Legionella is a serious phenomenon, particularly in the storage and distribution of hot water. Information can be found in SIA 385/1 [83] and the "SVGW Merkblatt" [88]. The following explanations have been summarized from [69].

The most well-known species of Legionella is the Legionella pneumophila, which is also the main cause of most Legionella diseases. However, these bacteria are only dangerous if aerosols contaminated with Legionella are inhaled in the form of the finest water droplets of about 5 µm in diameter and enter the lungs. However, drinking water contaminated with legionella is harmless.

Legionella reproduction occurs mainly at temperatures ranging from 25 °C to 45 °C. Legionella begins to die from 55 °C (Table 8.8). Below 20 °C they survive but are not able to reproduce.

Table 8.8 Survival time of Legionella [69].

Temperature °C	Time min	Effect
55.0	19	Reduction of the legionella numbers by one potency of 10 each (D-Value).
57.5	6	
60.0	2	
70.0	Some Seconds	

To prevent legionella multiplication the following principles are to keep in mind:

- The hot water distribution system must not have an unused pipe filled with water. This applies in particular to the subsequent shutdown of a tapping point.
- Drinking water that is not used at temperatures between 25 °C and 50 °C for more than 24 hours must be thermally disinfected, i.e. heated to 60 °C for one hour. This is only recommended in low-risk buildings.
- For instantaneous water heaters, the minimum temperatures do not apply if the hot water in the connected hot water distribution system does not remain at a temperature of 25 °C to 50 °C for more than 24 hours before being drawn off.
- The amount of stored hot water should be kept at a minimum (no oversized hot water heater storage devices).
- Tanks with hot water must be cleaned regularly, in particular descaled. Rust, lime and other deposits promote the formation of biofilms and serve as carriers for the growth of Legionella and other bacteria.
- Cold water pipes should be installed in such a way that heating by parallel hot water or heating pipes is avoided and a cold-water temperature of 20 °C maximum is achieved.
- Rarely used discharge points should be rinsed regularly.

The water temperature in a storage device utilized as preheating does not reach more than 45 °C due to the system and is therefore in the ideal range for Legionella reproduction. There should be as little as possible heated drinking water within this temperature range. Therefore, the standby volume of the storage in the area of the reheating of the drinking water should never fall below 60 °C. The standby volume is the expected peak volume (usually the highest peak of the hour) and corresponds to the volume that reaches the switch-off point of the storage charge. The reheating of the drinking water can be done with a secondary energy source, which can reach at least 60 °C.

In order to prevent the cooling of the hot water pipe system during downtimes, the circulation pump or the heating band should run in continuous operation. However, this leads to increased heat losses and can interfere with the stratification in the hot water storage tank.

When mixed-in-plants are used, supply and discharge lines with drinking water temperatures are operated at temperatures ideal for the reproduction of Legionella (25 °C to 45 °C). For this reason, these plants are not suitable for risk groups 1 and 2 without special measures (e.g. ionization or ozonisation). For risk group 3, it is recommended to heat the hot water in the installation once a day for at least one hour at 60 °C (Table 8.9).

Table 8.9 Risk groups for buildings and facilities and their measures for the legionella problem [69].

Risk class	Description	Building Category	Recommended treatment from the SVGW
1 High Risk	Buildings where people with weakened immune systems stay. Buildings with extensive pipe systems and installations with irregular water withdrawal (long stagnation times).	Hospitals with intensive care units, transplant departments or special departments (oncology, neonatology).	Follow the instructions compiled by those responsible for hygiene. Routine hot water temperature control and bacteriological analyses.
2 Medium Risk	In these buildings, risks result primarily from extensive installations, some of which involve long periods of stagnation.	Domestic multifamily houses MFH with central supply of domestic hot water, schools with showers, hotels, barracks, prisons, hospitals without the above-mentioned departments, retirement and nursing homes, sports buildings, indoor and outdoor swimming pools.	Compliance with the hot water temperature: at least 60 °C for one hour per day in the entire treatment plant and at least 50 °C at the taps. Additional measures must be taken in case of illness and positive results of the water analysis.
3 Minor Risk	Buildings with predominantly long stagnation phases.	Domestic SFH, Domestic MFH without hot water central supply, Administration, Schools without showers, Sales, Restaurants, Meeting rooms, Warehouses	If there are doubts about the hygiene of drinking water installations, appropriate investigations can be carried out. Measures must be taken if cases of illness occur and the water analysis is positive.

8.4 Hydraulic System

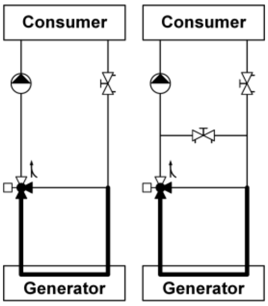
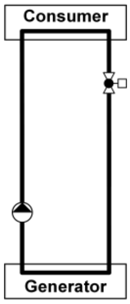
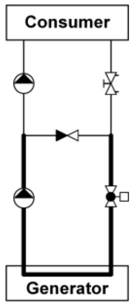
8.4.1 Hydraulic basic concept

The heat transfer can be arranged indirectly over a heat exchanger or directly (see also chapter 5.2). Indirect heat transfer is preferred nowadays. In order to achieve the lowest possible return supply temperature, the secondary side of the consumer's installation and the domestic heat distribution system must not have any hydraulic short circuits, between the supply and the return flow.

The following installations are therefore prohibited:

- Open expansion vessels
- Double distributor (pipe in pipe, square)
- By-passes (on distributors, consumer´s installations, etc.)
- Overflow regulators and valves between supply and return flow
- Injection circuits with three-way valves
- Deflection circuits with three-way valves
- Four-way mixer
- Heating coil / ventilation systems without zone valve.

Table 8.10 Admixing circuit, injection circuit with straight-way valve and throttling circuit.

	Admixing circuit	**Injection circuit with straight-way valve**	**Throttling circuit**
			
Conditions	• Low return temperature at low load (well suited for condensing heat generators) • Constant mass flow with variable supply temperature in the consumer circuit, provided that no mass flow controllers (e.g. thermostatic valves) are used for heat dissipation. • Variable mass flow over the generator circuit (regulated pump). • Uniform temperature distribution over the heating circuit.	• Low return temperature • Constant mass flow with variable supply temperature in the consumer circuit, provided that no mass flow controllers (e.g. thermostatic valves) are used for heat dissipation. • Variable mass flow over the generator circuit (regulated pump) • Uniform temperature distribution over the heating circuit • A main pump must always be taken into account for injection circuits with through control element.	• Low return temperature • Variable mass flow over the consumer and the generator circuit • Slow flow velocities at partial load • Uniform temperature distribution over the heating circuit • A main pump must always be considered for throttling circuits with straight-way valve.
Application	• Control of radiator and floor heating systems • Systems with low-temperature heat generators (condensing heat generation; condensing boilers) or heat pumps	• District heating connections (direct / indirect) • Control of radiator and floor heating systems • Systems with low-temperature heat generators (condensing heat generation; condensing boilers) or heat pumps • Domestic hot water heating with maximum temperature limitation for hard water or max. return temperature limitation • Air heater (heating coil) with risk of freezing	• District heating connections (indirect) • Thermal energy storage • District control of radiator and underfloor heating systems • Individual room control (e.g. via thermostatic valves) • Cooling Registers in large systems • Air heater (heating coil) • Plants with low-temperature heat generators (condensing heat generation; condensing boilers)
Advantages	• Low return temperatures • Good controllability when used on unpressurized or low-pressure manifolds • Several heating circuits influence each other less (simpler adjustment, more stable)	• Low return temperatures	• Low return temperatures • Simple individual room control via thermostatic valves
Disadvantages	• no differential pressure permitted on the primary side (low-pressure distributor)	• Use of at least two circulating pumps • Manifold must have differential pressure	• Danger of freezing with air heaters (register) • Manifold must have differential pressure

8.4.2 Basic hydraulic circuits

Practically all circuits in building design technology are based on the following basic hydraulic circuits:

- Admixing circuit
- Deflecting circuit
- Injection circuit
- Throttling circuit

In the following designs, only the admixing circuit and the injection circuit with straight-way valve and the throttling circuit are discussed (Table 8.10). These are the most frequently used circuits in building technology today. The injection circuit with straight-way valve and the throttling circuit are used as basic hydraulic circuits for the primary-sided district heating connection.

8.4.2.1 Admixing circuit

Heating groups on the secondary side (low-pressure distributor) are preferably designed as admixing circuits. The return water is mixed with the supply water to the desired supply temperature of the heating group. The desired operating medium temperature of the heating group is defined by the heating curve and depends on the outside temperature.

When using thermostat valves, the mass flow in the heat dissipation circuit becomes variable, which is why it is mandatory to use a controlled circulating pump. If the primary operating medium temperature delivered by the district heating supplier is higher than the required secondary operating medium temperature in the heating group (e.g. underfloor heating), an internal bypass must be integrated at the admixing circuit of the secondary side.

In general, an internal bypass is necessary if the following temperature ratio (supply distribution - return heating circuit) / (supply heating circuit - return heating circuit), is greater than 2.5. Due to the continuous addition of cooled return water via bypass, the operating medium temperature decreases and thus the entire valve stroke is available to the control organ for regulation. This makes optimum use of the control capacity of the adjusting organ.

8.4.2.2 Injection circuit with straight-way valve

In the case of differential pressure-affected distributors, a pressure difference is provided to the consumer circuit by the pump in the generator circuit (e.g. district heating pump in the primary circuit). Depending on the position of the straight-way valve, more or less supply water from the heat supplier (heat generator, primary circuit) is injected. This results in a temperature control with constant flow on the consumer circuit and a quantity control with variable flow rate. For this reason, a speed-controlled network pump should be used.

For direct connections to a district heating network, an injection circuit with a straight-way valve is therefore preferably used. This prevents a high return supply temperature in partial load operation. A further area of application is hot water heating. In the case of hard water, calcification of the secondary side of the heat exchanger due to a too high operating medium temperature on the primary side can be avoided.

8.4.2.3 Throttling circuit

Like the injection circuit, the throttling circuit is a hydraulic circuit subject to differential pressure and is used, for example, for indirect district heating connections on the primary side.

In contrast to the injection circuit with straight-way valve, the consumer circuit has a variable flow rate control. For this reason, a speed-controlled network pump should also be utilized.

8.4.3 Control Valve

Control valves of different designs are used in all basic hydraulic circuits:

- Straight-way valves with one inlet and one outlet
- Three-way valves with two inlets and one outlet (mixing valves) or one inlet and two outlets (distribution valves, rarely used).

In order to determine the pressure loss of control valves and to design the hydraulic system, the flow characteristic open valve k_{VS} and the flow characteristic valve k_V value must be initially determined.

In the case of control valves and generally with adjustable fittings, the pressure loss is not calculated with individual resistances, as with standard fittings or pipe fittings, but with the so-called k_{VS} value. The manufacturers of the control valves therefore specify a k_{VS} value for each control valve. This corresponds to a volumetric flow rate in m^3/h between inlet and outlet of the fully open control valve at 1 bar pressure loss (Δp_{V100} = 1 bar). This relationship applies to cold water at a temperature between 5 °C and 30 °C. With increasing temperature, the density of water decreases, water at 100 °C has a density about 4% lower than at 20 °C; this must be taken into account at high water temperatures.

The k_{VS} value can be used to determine the pressure loss in the event of a failure with the control valve fully open and, conversely, to correctly design the control valve.

$$k_{VS} = \frac{\dot{V}_{max}}{\sqrt{\Delta p_{V100}}}$$

$$\Delta p_{V100} = \left(\frac{\dot{V}_{max}}{k_{VS}}\right)$$

max relates to the maximum volumetric flow rate by design.

If the valve is set to a certain stroke (stroke between 0 % - 100 %), the **k_V-value** of a valve indicates the volumetric flow rate in m^3/h, that is obtained when the pressure difference between the inlet and outlet of a valve is 1 bar (partial load).

$$k_V = \frac{\dot{V}}{\sqrt{\Delta p}}$$

The hydraulic behavior of the control valves is described by the so-called basic characteristic curve (stroke in function of the flow). Both straight-way and three-way valves are usually offered with two different basic characteristics curves:

- Linear characteristic curve: same variation of stroke result in same flow rate changes.
- Equal-percentage characteristic curve: equal variation of stroke result in an equal percentage change in the current flow rate.

Figure 8.2 shows a linear and an equal percentage characteristic control valve curve. When the valve is open (i.e. maximum stroke H_{100}), the largest k_V value is reached, which is called the k_{VS} value. When opening the valve (no stroke), there is a volume surge of approximately 1% to 5% of k_{VS}. Valves should therefore be designed tightly so that there is no need to regulate in this area.

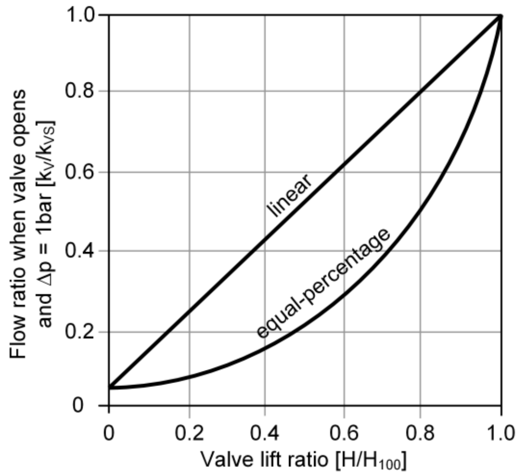


Figure 8.2 Characteristic curve of control valves (linear and equal-percentage characteristic curve)

With a **linear characteristic curve**, the same changes in the k_V value result in the same changes in the valve stroke ΔH. With an **equal-percentage characteristic curve** the same percentage changes in the k_V value are made with the same percentage alliterations in the valve stroke ΔH.

In the design of the valves, in addition to the valve authority, the heat transfer behavior of any heat exchangers in the control circuit plays a role, which is expressed by the heat exchanger characteristic value a (a-value). Since the calculation of the “a value” is relatively complex, this is usually taken into account by the appropriate choice of the characteristic curve of the control valve. This simplification is generally permitted as long as a valve authority of ≥ 0.5 is adhered to. The following simplified rule for the choice of the characteristic curve applies:

- Control loop without heat exchanger = Control valve with linear characteristic curve
- Control loop with heat exchanger = Control valve with equal-percentage characteristic curve.

8.4.4 Valve authority

When a valve is installed in a hydraulic circuit, it no longer behaves according to the characteristic curve, because the pressure difference above the valve becomes a variable part of the overall pressure gradient of the system. As a result, the characteristic curve is more or less deformed. As deformation increases, control accuracy and speed are increasingly impaired. In extreme cases, the control circuit becomes unstable and begins to oscillate. The value for the deformation of the characteristic curve is the valve authority, which is defined as follows:

> The **valve authority N** is the ratio between the pressure difference above the control cross-section of the opened valve at nominal discharge rate Δp_{V100} and the maximum pressure difference occurring above the control cross-section of the closed valve when it is about to open Δp_{V0}.

The pressure drop above the part of the circuit whose variable flow is influenced by the valve plays an important role here (presented in bold lines in Table 8.10).

In circuits with **three-way valves** there are no problems as long as this condition applies:

$$N = \frac{\Delta p_{V100}}{\Delta p_{V100} + \Delta p_{var100}} \geq 0.5$$

At 100 % flow, the pressure drop across the open three-way valve (Δp_{V100}) must be equal to or greater than the pressure drop across the route with variable flow rate (Δp_{var100}): $\mathbf{\Delta p_{V100} \geq \Delta p_{var100}}$.

There are no problems when switching with **straight-way valves**, as long as this applies:

$$N = \frac{\Delta p_{V100}}{\Delta p_{V0}} \geq 0.3$$

A valve authority $N_V \geq 0.5$ is strictly speaking also considered as a target value here, but in the case of straight-way valves, the limit value $N_V \geq 0.3$ often has to be used. The pressure drop above the open straight-way valve at 100% flow rate (Δp_{V100}) must be at least 30% of the maximum pressure difference (or at least 3000 Pa) above the closed valve when it is just beginning to open (Δp_{V0}):

$\Delta p_{V100} \geq 0.3\ \Delta p_{V0}$

As the necessary pressure drop Δp_{V100} results from Δp_{V0}, the problem is reduced to the question of what should be used as the maximum occurring pressure difference across the closed valve Δp_{V0}. In the worst case (unregulated pump) this is the maximum delivery head at a zero flow rate, i.e. a very high value, especially for pumps with steep characteristic curves. Significantly lower values result if the following is observed:

- Exactly define the operating point that is assumed to be the worst case for the maximum pressure difference. (also cumulative to the other possibilities).
- Utilize the district heating pump with constant pressure control. This results in a completely flat pump characteristic curve (see also chapter 3.3.3).

- Provide a measuring point for the differential pressure control of the pipeline pump within the heating network instead of at the pump (see also chapter 3.5).
- Utilize a district heating pump with proportional pressure control (see also chapter 3.3.3).
- Install automated differential pressure regulators at the customer. The set point of such controllers can usually be between 10 and 100 kPa. Although this allows reasonable control valve designs to be realised without any problems, larger connection pressure differences are necessary (see chapter 8.4.5.2).
- Install pressure-independent control valves. In this type of construction, differential pressure regulators are used in the same casing without auxiliary energy and control valve. The differential pressure measurement for the hydraulic regulation of the valve is carried out directly above the control valve (see chapter 8.4.5.3).

Attention:
Often several groups are connected as admixing circuits with low pressure difference. This means that each district heating pump receives water through the valve and the route with variable flow. In this context, the question arises: "How large can the pressure difference over the variable flow distance be?" If this pressure difference becomes too large, the individual groups (district heating pumps) influence each other. In order to prevent this - in addition to the rule on valve authority - the following rule must be adhered to:

If several groups of low differential pressure are connected as admixing circuits, the maximum differential pressure over the section with variable flow must not exceed 20 % of the delivery head of the smallest district heating pump at this design point.

8.4.4.1 Automated differential pressure regulator

Nowadays, automatic differential pressure regulators are often installed in heat networks at the customer. The set point of such controls can usually be set between 10 and 100 kPa. This makes it possible to implement efficient control valve designs without any problems, but larger connection pressure differences are also necessary. The design of the differential pressure regulators and control valves depends on the extent of the pressure difference fluctuations in the heat network. Decisive for the pipe network calculation is the route to the most decisive, usually the most remote customer. Therefore, the following recommendations:

- The use of differential pressure regulators without auxiliary energy makes sense if clearly defined pressure conditions are to prevail at each customer.
- Design the heat exchanger for the most remote customer with the smallest possible pressure drop. Design the pressure drop across the control valve according to a valve authority of 0.3...0.5.
- Realistically calculate the maximum possible pressure difference across the almost closed differential pressure regulator that is just starting to open:
    - Use the theoretical possible maximum value for minor heating networks in order to calculate the valve authority and realise a sufficient valve authority ≥ 0,2 until 0,3.
    - In extended heating networks, where many connected customers together result in a more or less uniform weather-related load, a lower value for the maximum pressure difference occurring can be expected (e.g. 50%) or a correspondingly lower valve authority (e.g. ≥ 0,1 up to 0,15) can be accepted.
- For customers located nearby, provide sufficient pressure drop above the open differential pressure regulator (design case), where a sufficiently large connection pressure difference is always available.
- In systems with pressure difference measurement in the network or with pumps with proportional pressure control, it can happen that customers at the beginning of the network suddenly have too little pressure difference at low load of the overall system. Here it makes sense not to set the valve authority unnecessarily high.

8.4.4.2 Pressure independent control valve (PICV)

For the pressure independent control valve (PICV), the differential pressure regulator without auxiliary energy and the control valve are realised in one housing (Figure 8.4). The differential pressure measurement is carried out directly above the control valve. This means that the valve authority of the control valve is only dependent on the P-band of the differential pressure controller. The total pressure drop across the PICV is thus made up as follows:

Δp PICV = Δp Control valve + Δp Differential pressure regulator

The pressure difference above the control valve is fixed by the manufacturer (e.g. 20 kPa), and the pressure drop above the pressure differential regulator is given by its k_{VS} value. Here, too, a minimum valve authority must be adhered to with regard to the pressure differential regulator.

PICV’s have the advantage that pressure differential regulator and control valve are conveniently housed in a single housing. Otherwise, the above versions of the automatic pressure differential regulator apply with the difference that the pressure difference measurement is carried out directly above the control valve.

8.4.5 Variable flow systems

Energy-efficient hydraulic systems are designed and built in such a way that they have a variable mass flow. This means that the mass flow reacts to the operating conditions and subsequently the circulating pump adjusts the speed. This chapter explains the behavior of the hydraulic system when consumers open and close the control valves. Details are executed in [70] and summarized below.

8.4.5.1 Static hydraulic balancing

In static hydraulic system balancing there is a division of tasks between:

- the **control valve** which is oriented towards control technology and
- the **balancing valve** which is responsible for the static alignment of the hydraulic system.

The static balancing valve (sectioning control valve) is used for manual adjustment of hydraulic systems. The valves have an integrated pre-setting function for the precise determination of the flow rate. Sectioning control valves have fixed pre-settings. The characteristics of static alignment are:

- Static hydraulic alignment causes undersupply and oversupply of consumers in cold and heat supply systems.
- In the partial load range, the different mass flows rates influence the return supply temperatures of the heat generator.
- High oversupply can cause an increase (high mass flow rate, low ΔT) and undersupply (low mass flow rate, high ΔT) and a drop in the return temperature.
- Depending on the hydraulic system integration of the heat generator, this can influence the efficiency or the operating medium temperature.
- The networks can be designed to be more balanced in partial load cases if the pumps are regulated to pressure = constant. However, this measure is at the expense of energy efficiency and the power requirement for the circulating pump.

Because of the basic disadvantages, the type of hydraulic system with static balancing and regulation is no longer up-to-date, either for district heating networks, or for internal heat distribution.

8.4.5.2 Hydraulic balancing with automatic differential pressure regulator

The following task division exists for hydraulic alignment with an automatic differential pressure control:

- the **control valve** with a specific control-based task and
- a **dynamic balancing valve** responsible for hydraulic system alignment in all operating modes
- with each automatic differential pressure valve, the pressure above the consumer and the control valve is kept constant in all operating modes (partial load operation).

In hydraulic systems, the pressure ratios between individual consumers and strands can vary greatly depending on the power requirement of each consumer. It is therefore necessary to produce uniform pressure conditions in the individual sections of the plant, irrespective of the needs of the individual consumers. In the primary circuit of a district heating network, the substation is an example of such a consumer. Examples in the secondary circuit are climbing strands, radiators, ceiling cooling surfaces, underfloor heating distributors, fan coils, induction devices or air heaters.

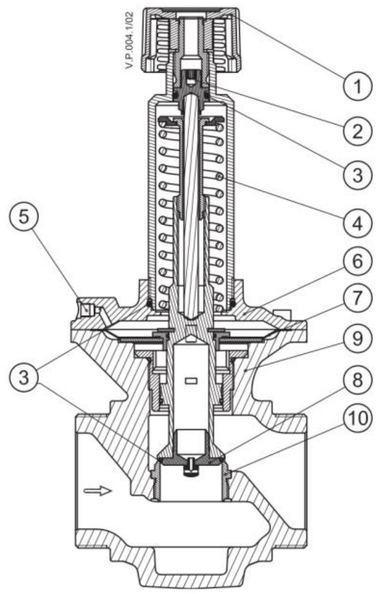


Figure 8.3 Section of an automated differential pressure regulator ASV (Danfoss).

Automatic differential pressure regulators ensure a constantly adjustable differential pressure over a section of the plant. Via an internal connection and together with the set point spring, the pressure in the return flow acts on the underside of the control diaphragm (Figure 8.3; 7), while the flow pressure acts on the diaphragm from above via an impulse line (Figure 8.3; 5). With the set spring force, the set differential pressure is kept constant by the differential pressure valve in the subsequent system section. Automated differential pressure controls must be implemented in the return flow, which are combined with a partner valve (sectioning control valve) in the supply. Automated differential pressure regulators limit the differential pressure not only under design conditions, but also under partial load.

By controlling the pressure in the partial load case, flow noise in fittings can be avoided. The control of differential pressure allows the subsequent control valves, for example thermostat valves, a higher valve authority and thus a more precise and stable temperature control and energy saving.

The most important characteristics of hydraulic alignment with automatic differential pressure regulators are:

- The alignment causes practically no under or oversupply.
- The automatic differential pressure control takes over the hydraulic alignment during partial load operation and reacts to changes in the mass flow rate.
- The control of the heat exchangers and district heating pumps within the system has to tolerate any changes in the hydraulic system maintaining stability and energy efficiency.
- The different mass flow rates have a small influence on the return temperatures and thus on the heat generator.

- The efficiency or the operating medium temperature of the heat generators is not affected by incorrect hydraulic alignment in partial load operation.
- The characteristic pump control curve is of great importance in order to avoid low pressure during partial load operation and thus leaving the consumer undersupplied.

8.4.5.3 Hydraulic balancing with pressure independent control valve

Using hydraulic balancing with pressure independent control valve (PICV), the division of functions between the control valve and a static or automated balancing valve is no longer considered. In this system the control and hydraulic functions are integrated in a dynamic valve. With the PICV, the pressure above the consumer and the control valve are kept constant with one valve in every operation (partial load operation).

The PICV (Figure 8.4) consists of a differential pressure regulator and a control valve.

The differential pressure control holds a constant differential pressure above the control valve. The force of the spring counteracts the differential pressure of p_{CV} (p2-p3) on the membrane. If the differential pressure changes above the control valve (because the available pressure changes or due to an action of the control valve), the hollow cone shifts to a new position, which leads to a rebalancing and thus the differential pressure maintains a constant level.

The control valve has a linear characteristic. It has a preset value k_V in the form of a maximum limit of the valve stroke. The percentage shown on the scale corresponds to the percentage of the flow rate. The PICV shown in Figure 8.4 can be used to change the setting by raising the pre-setting and turning the upper part of the valve to the desired position (the percentage displayed on the scale). A locking mechanism prevents the valve from being inadvertently dislocated.

As described above, the pressure difference is kept constant via the control valve. The pressure loss above the control valve is therefore constant. As a result, a constant valve authority N = 1.0 can be assumed. The pressure drop over the variable part of the section is not relevant because the pressure drop is kept constant via the control valve.

The most important features of hydraulic system alignment with pressure independent control valves are:

- The alignment causes practically no under or oversupply to the consumers.
- The differential pressure-independent control valve performs hydraulic system alignment in partial load operation and reacts to changes in the mass flow rate.
- The control of the heat exchangers or district heating pumps in the heat delivery systems does not have to react to the hydraulic system changes and can be regulated with more stability and energy efficiency.
- Different mass flow rates have no negative influence on the return temperatures and thus on the heat generator.
- The efficiency or the operating medium temperature of the heat generators is not affected by incorrect hydraulic system alignment in partial load operation.
- The characteristic pump control curve must be observed in order to prevent a pressure drop for consumers with high cooling and heat demands during partial load operation.
- The technical data of devices from all manufacturers should be verified and published by an independent measuring institute.

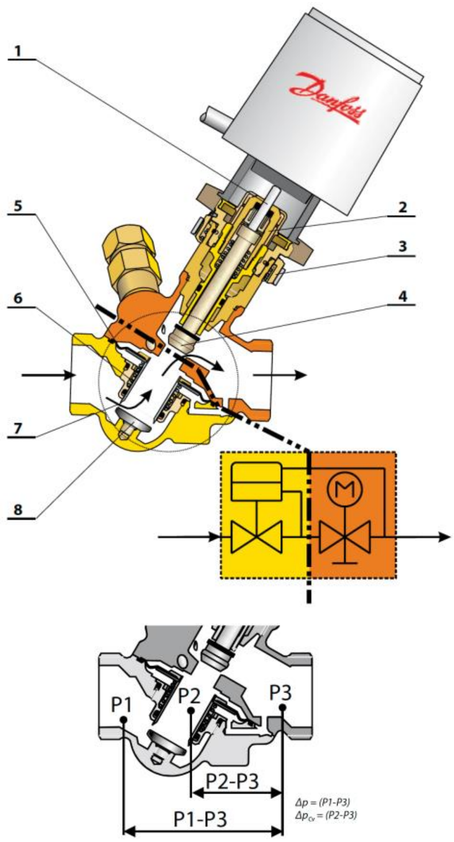


Figure 8.4 Section through a pressure independent control valve (AB-QM from Danfoss). The valve consists of a differential pressure regulator (yellow) and a control valve (orange).

8.4.5.4 Comparison of hydraulic balancing systems

Over and under supply

In systems with automated differential pressure regulator or PICV there is practically no undersupply or oversupply. Therefore, the temperature difference and as a result also the return temperature in partial load operation does not differ from the set point. The heat generators can be operated without loss of energy efficiency.

Pump pressure

The systems with automatic differential pressure regulator or PICV require roughly the same pressures and electrical power consumption of the pumps. Thus, both systems can be equipped with approximately the same energy-efficient pumps. Systems with static balancing require much higher pump power and thus electricity. Improper hydraulic system design therefore leads to high power consumption over years or decades.

Characteristic pump control curve (Table 8.11)

Systems with static balancing can usually be operated with constantly controlled characteristic pump control curves. This will prevent an under supply for consumers. This hydraulic system should no longer be used in modern systems that handle variable mass flow rates.

Systems with automatic differential pressure regulator can usually be operated with proportionally controlled pump control characteristic curves, whereby the minimum conveying pressure in the partial load range must be sufficiently observed by all consumers.

Systems with a PICV can usually be operated with proportionally controlled characteristic pump control curves, whereby the minimum pressure in the partial load range must be sufficiently observed by all consumers.

Volumetric flow rate by partial load

The system with PICV requires a smaller volumetric flow rate in partial load operation, because the valve authority is higher. Thus, due to the reduced required volumetric flow rate, electrical pumping energy can be saved.

Table 8.11 Characteristic pump control curves for the various control and hydraulic balancing systems. *Observe minimum delivery pressure in partial load case.

Hydraulic system balancing	Characteristic Pump control curve
Static comparison	constant
Automatic differential pressure control	proportional*
Differential pressure independent control valve	proportional*

Systems with static balancing are as hydraulic system inefficient and no longer used in modern variable mass flow rate systems.

For hydraulic systems in which fittings with a minimum pressure difference are installed, it is necessary to determine whether these pressure differences must also be observed in partial load operation. This has a decisive influence on the course of the pump control characteristic.

Partial systems of an entire system can also operate in partial load operation of the entire plant near the set point and thus almost require the output volume flow. Thus, if the pump control characteristic curve is too low, all hydraulic systems may experience undersupply.

Increased attention must be paid to the choice of the characteristic pump control curve in various hydraulic systems.

8.5 Standard-Circuits

An overview of the connection possibilities of district heating connections can be seen in Figure 8.5. It is subdivided as follows:

- Direct or indirect connection
- With or without domestic hot water heating
- Type of hot water heating (instantaneous water heater and hot water storage with internal or external heat exchanger)
- Connection of domestic hot water heating to the primary or secondary circuit.

Chapter 5.2 describes in more detail the basics of direct and indirect connections. On the following pages, the most important standard circuits for direct and indirect district heating connections are schematically presented and briefly described. In the schematic diagram, a pressure independent control valve (PICV) is used throughout for volumetric flow rate and differential pressure control. Alternatively, the volumetric flow rate and differential pressure can be controlled via a separate straight-way valve for volumetric flow rate control and a differential pressure regulator.

A detailed representation of the measuring and control technology (e.g. controllers and active lines) has been omitted for the sake of simplicity.

Different standard circuits for hot water heating are shown. In the diagrams, the two hot water tanks with internal or external heat exchangers are considered in more detail. The hot water tanks are charged via a hot water priority circuit (obsolete: boiler priority circuit). Alternatively, the charge can also be carried out in parallel. The parallel charge of the hot water tanks is more complex and cost-intensive in terms of regulation. The connection load must be designed accordingly, but has advantages in the case of increased hot water requirements (e.g. hospital or retirement home).

In addition, two circuits with jet pumps are also listed.

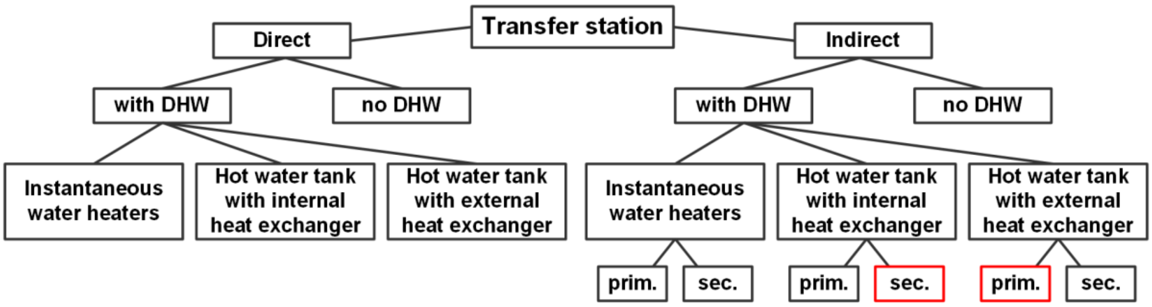


Figure 8.5 Connection options for heat supply with or without domestic hot water heating. A red frame indicates the usual type of connection for domestic hot water heating with an indirect connection. Prim. refers to the connection of domestic hot water heating to the primary and sec. to the connection to the secondary side.

8.5.1 Direct connection

The heating circuits have a weather-compensated supply temperature control and the primary return temperature is limited by a return supply temperature limit if the maximum permissible return temperature is exceeded.

Direct connection of a heating group by means of a **throttle circuit** according to Figure 8.6 (variable flow in the heating group, e.g. connection of an air heater).

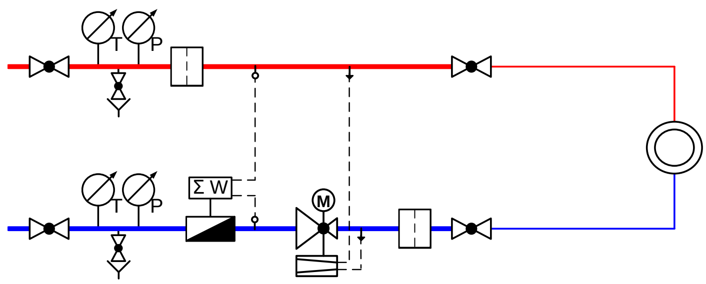


Figure 8.6 Direct connection with throttle circuit.

Direct connection of a heating group by means of an **injection circuit** for temperature control according to Figure 8.7 (e.g. connection of a radiator or floor heating).

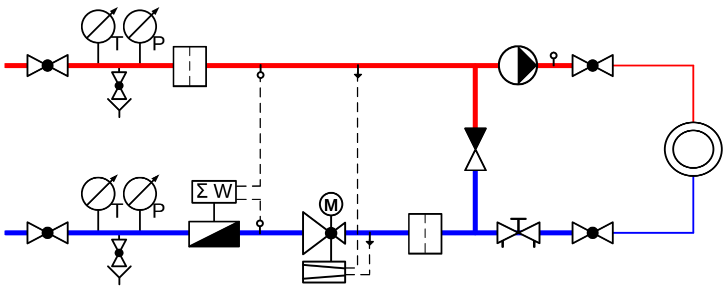


Figure 8.7 Direct connection with injection circuit.

8.5.2 Indirect connection

The connection to a district heating network must be made indirectly in the case of large geodetic differences in height and/or extended systems, as the network pressure exceeds the maximum permissible pressure in the domestic heat distribution system (e.g. domestic heating). This means that the generator circuit is as hydraulic system separated from the consumer circuit by a heat exchanger (primary and secondary circuit). The heating circuits have a weather-compensated control of the operating medium temperature on the secondary side and via the straight-way valve on the primary side (pressure independent control valve).

The **indirect connection** to a district heating network is usually carried out with a throttle circuit as in Figure 8.8. The primary return temperature shall be limited with a return temperature limit if the maximum permissible return temperature is exceeded.

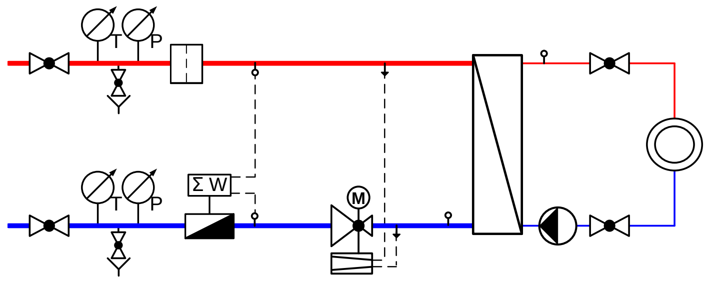


Figure 8.8 Indirect connection with a throttling circuit.

Indirect connection with a primary-sided **heat exchanger** for several **heating groups** and **hot water heaters**. The secondary-sided control is carried out in accordance with the explanations in Chapter 8.4. The domestic hot water heating is presented in a simplified way – the execution should be carried out according to the explanations in the next chapter.

The circuit presented by Figure 8.9 is only possible if the secondary-side connection of the heat exchanger has a low pressure difference and that the following frame conditions are met:

- Valve authority ≥ 0.5, i.e. the pressure drop across the three-way valve is greater than the pressure drop across the section with variable flow rate (corresponds to heat exchanger and connecting pipes).
- Maximum pressure drop across the secondary side sections with variable flow ≤ 20 % of the delivery head of the smallest district heating pump. This prevents the influence of erroneous circulation on different heating groups with three-way valves.

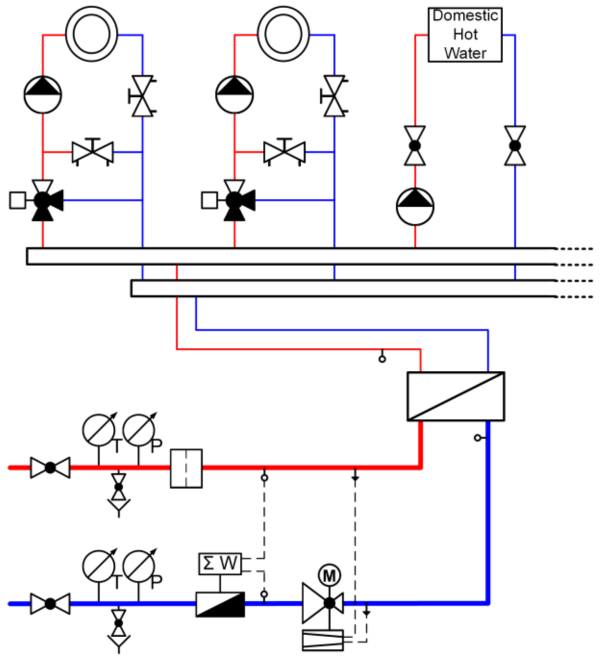


Figure 8.9 Indirect connection with throttle circuit for secondary connection of several heating groups and hot water heaters.

Indirect connection via a **decentralized thermal energy storage** (Figure 8.10), e.g. for several **heating groups** and **hot water heaters** or for further **sub-distribution**. The on/off criteria for charging the thermal energy storage are recorded using temperature sensors at the top and bottom of the heat storage device. The hot water heating is simplified her - the execution should take place according to the explanations in the next chapter.

On the secondary side, low-pressure difference connections must be provided. The secondary sided control is carried out as described in Chapter 8.4. This connection is suitable, for example, for customers with high peak loads.

Attention: With the direct connection of the thermal energy storage device, the nominal pressure of the storage device must be observed.

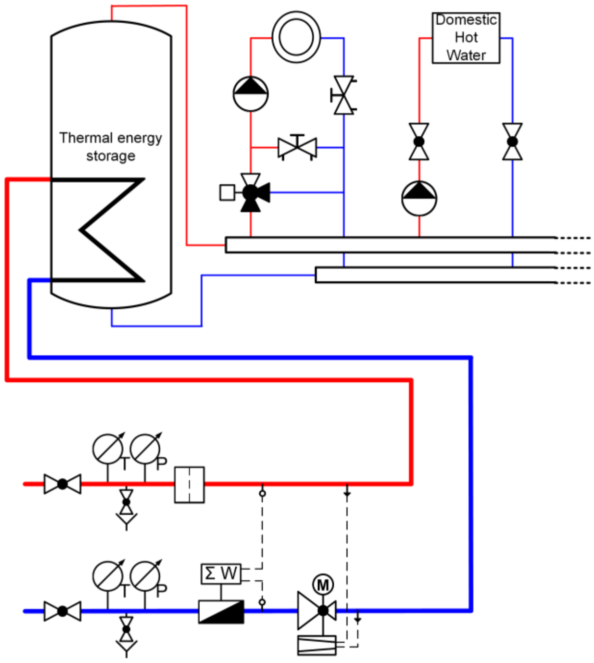


Figure 8.10 Connection of a decentralized thermal energy storage tank for indirect connection of several heating groups and hot water heaters.

8.5.3 Domestic hot water generation

The following standard circuits deal with domestic hot water heating. The hot water heaters must always be charged via a domestic hot water priority control. Alternatively, charging can also take place in parallel. Parallel charging of the domestic hot water tank is more difficult and cost-intensive in terms of control technology. Furthermore, the connected load must be designed accordingly, but has advantages in case of an increased hot water demand (e.g. hospital or retirement home).

Direct connection of **hot-water storage tank with internal heat exchanger**. The maximum permissible return supply temperature must be guaranteed by a suitable hydraulic system and control measures. The switch-on and switch-off criteria are recorded via temperature sensors in the storage device.

With the circuit shown in Figure 8.11, the hot water storage tank with internal heat exchanger is charged directly and at a constant flow rate.

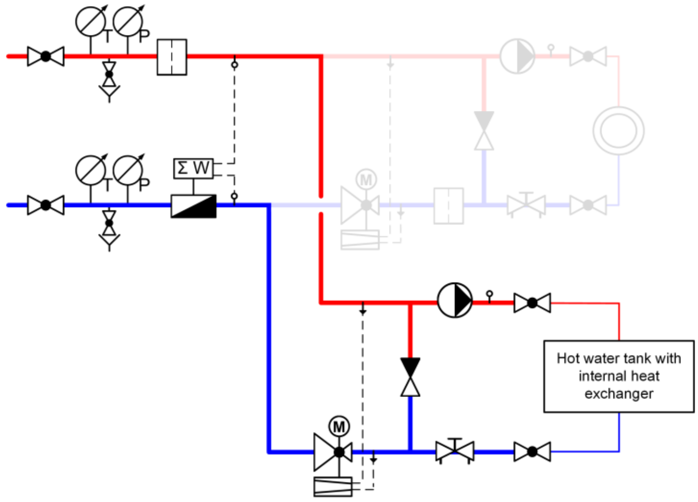


Figure 8.11 Direct connection of a hot-water storage tank with internal heat exchanger.

Connection of a hot water storage tank with external heat exchanger. Provides constant high heating output with constant high hot water temperature and defined low return supply temperature.

The same circuit also applies to an indirect connection of a hot water storage device with internal heat exchanger. The maximum reliable return supply temperature must be guaranteed by a suitable hydraulic system and control measures.

The charging control of the hot water storage device is carried out by a single injection circuit for temperature control with constant flow rate. This prevents calcification in the heat exchanger due to increased water hardness of the drinking water by limiting the primary charging temperature of the heat exchanger to 65 °C and 70 °C.

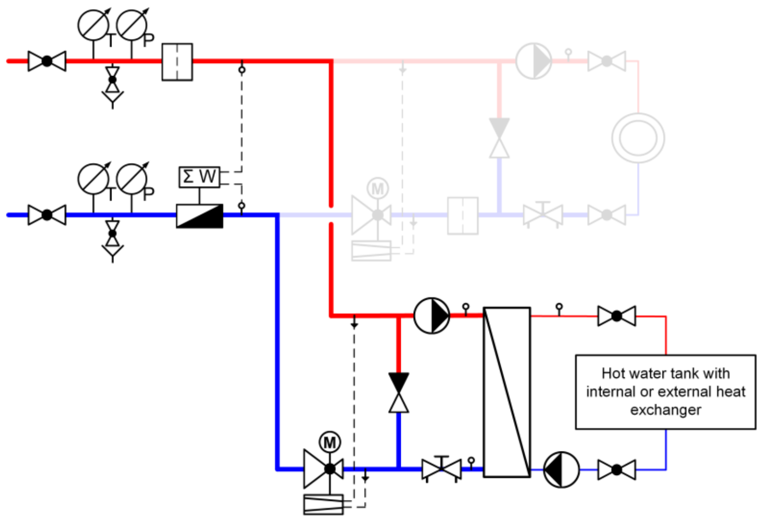


Figure 8.12 Connection of a hot water storage tank with external heat exchanger or an indirect connection of a hot water storage tank with internal heat exchanger.

The on/off criteria are determined by temperature sensors in the hot-water storage tank. With the formwork shown in in Figure 8.12, the secondary charging takes place with constant or variable flow rate. At variable flow rates, the speed-controlled pump is started up with a defined start-up ramp and controlled to the outlet temperature.

The pump must be temperature and pressure resistant.

Connection of a hot water storage tank with external heat exchanger. Provides consistently high heating output with consistently high hot water temperature.

With the admixing circuit on the secondary side shown in Figure 8.13, the hot water storage tank is charged at a constant flow rate. This allows the supply temperature on the secondary side to be controlled by the hot water heater, thus avoiding a supply temperature that is too low when the storage device charging starts. With increased water hardness in the drinking water, overheating in the heat exchanger can be prevented when the storage tank charge is switched off by cooling the heat exchanger over a certain follow-up time (e.g. 5 min.).

The pump must be temperature and pressure resistant.

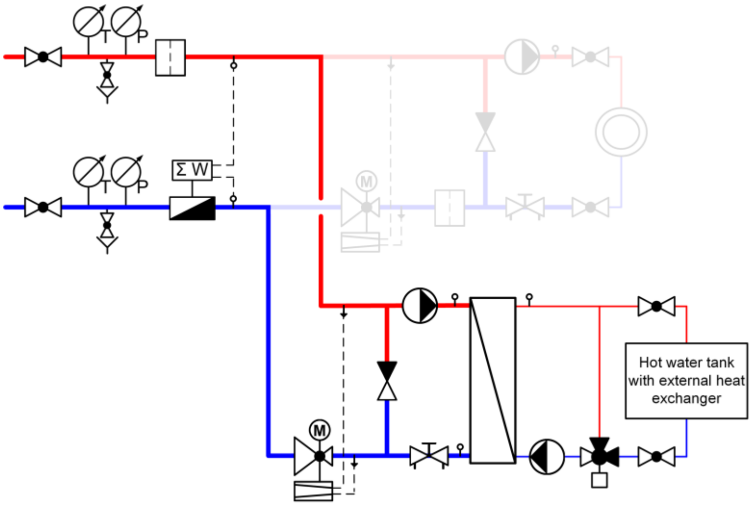


Figure 8.13 Connection of a hot water storage tank with external heat exchanger and on the secondary site the admixing circuit.

Low return temperatures are essential for the economical operation of district heating networks. The focus here is on the hydraulic system integration of the customers and there is potential for optimisation, especially for domestic hot water heating. For example, the return supply temperature can be further reduced by preheating the cold water via an additional heat exchanger (preheater) or via an additional sanitary sided storage device.

The principle of cold-water preheating with an additional heat exchanger can be applied to all types of hot water production (see Chapter 8.2). Figure 8.14 shows an example of a variant of cold water preheating for a hot-water storage tank with an external heat exchanger. The pump must be temperature and pressure resistant.

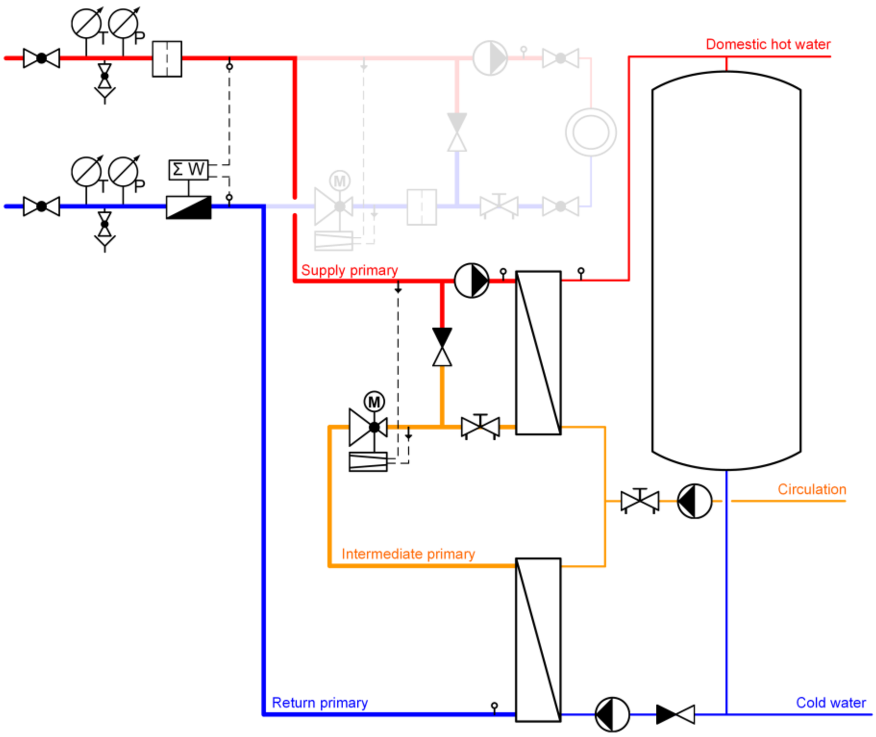


Figure 8.14 Connection of a hot-water storage tank with external heat exchanger and preheating of the cold water via a separate heat exchanger.

Another variant of preheating offers series-connected hot water storage tanks with internal heat exchanger, whereby the first hot water storage tank (right) is used for preheating and the downstream hot water storage tank (left) is reheated in order to reach temperatures eliminating Legionella (Figure 8.15).

The water temperature in a storage device dedicated to preheating usually does not reach more than 45 °C and therefore favours Legionella reproduction. Therefore, the hot water in the after heater should be heated up to more than 60 °C for at least one hour per day.

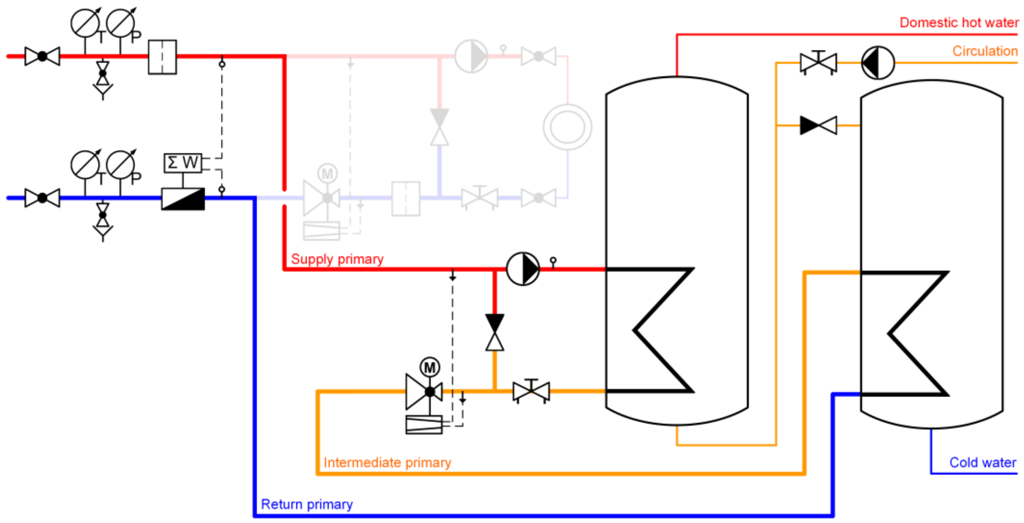


Figure 8.15 Connection of two in series connected hot water storage tanks with internal heat exchanger.

8.5.4 Jet pump

Another connection option is the jet pump connection. This circuit can be used as direct connection (Figure 8.16) or indirect connection (Figure 8.17) analogous to the circuits listed above.

The jet pump provides temperature control with a variable flow rate. The jet pump assumes the function of the volumetric flow and differential pressure control, which generally dispenses with the pressure independent control valve.

Comparison:

The throttling circuit shown in Figure 8.6 provides volume control with a variable flow rate. The injection circuit of Figure 8.7 presents temperature control with a constant flow rate.

Caution is required in poorly aligned heat networks with variable flow rates (risk of "death" at low load).

In order to minimise calcification in the heat exchanger by increased water hardness in drinking water, the primary charge temperature of the heat exchanger can be limited without the need to use a circulating pump and a check valve (see injection circuit).

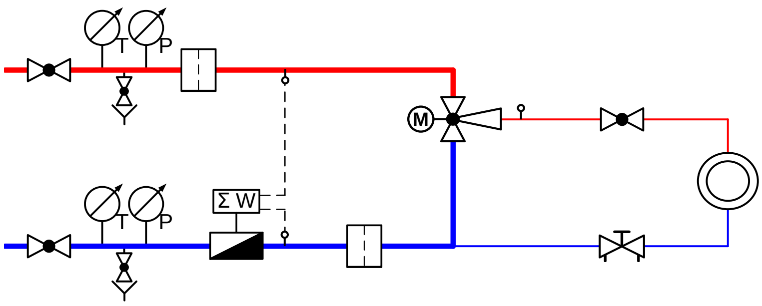


Figure 8.16 Direct connection with jet pump.

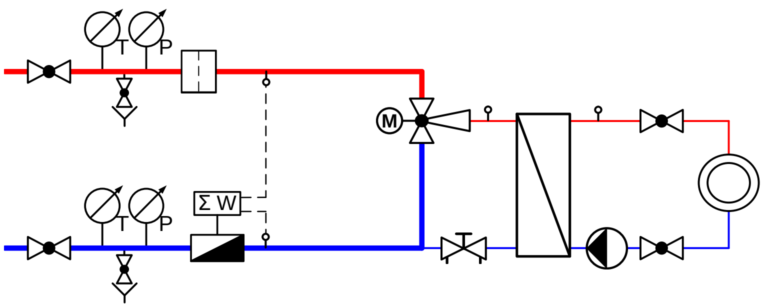


Figure 8.17 Indirect connection with jet pump

8.6 Requirements for domestic heat distribution

The following requirements concern building technology, heating system, ventilation system, domestic hot water heating and control technology, i.e. mainly the secondary side owned by the customer (domestic heat distribution). From the point of view of the heat supplier, however, there is a great deal of interest in the technical situation on the secondary side of the customer. The heat supplied should be used as efficiently as possible with the lowest possible primary return temperatures. From a legal point of view, however, there is no obligation on the heat supplier to act on the building technology of the customer, unless it has been contractually agreed (e.g. in the heat supply contract).

The following recommendations are therefore mainly dedicated to heat suppliers in regard to potential heat customers, which recommendations can be made to heat customers and which can possibly also be contractually agreed upon.

The recommendations listed here do not deal with process heat. However, the recommendations for process heat do not deviate fundamentally from those made here.

8.6.1 Building technology

- For the lowest possible (primary) return temperature, there must be no hydraulic system short circuit between the supply and return flow on the primary or the secondary side, which is why the following installations are not permitted:
    - Open expansion vessels
    - Double distributor (Pipe in pipe)
    - Bypasses (for distributors, for consumers etc.)
    - Overflow regulators and valves between supply and return
    - Injection circuits with three way-valves
    - Baffle switches with three-way valves
    - Four-way mixers
- New or modified connections must be designed in a way that they can be insulated as continuously as possible (ball valves with stem extensions, temperature and pressure indicators and filling and draining valves pulled out far enough).
- Removal or avoiding of unregulated bypasses and hydraulic switches between supply and return lines (e.g. keeping the supply line warm for ventilation systems).
- On the primary side, bypasses or overflow valves to ensure a minimum flow at the end of the lines (e.g. prevention of "cold legs") are only permissible if no other solution is possible and it can be guaranteed that the flow is minimal.
- To prevent the return temperature from rising in the event of erroneous circulation or if the flow rate on the primary side is greater than the flow rate on the secondary side (e.g. error in the straight-way valve), non-return valves are used in the bypasses of injection circuits on the primary side. However, the following disadvantages of a non-return valve must be taken into account when designing the circuit and an erroneous circulation must be avoided as far as possible:
    - One-sided hydraulic decoupling
    - Pressures are added in case of erroneous circulation
    - Group heats up despite the pump being switched off if the straight-way valve is opened unintentionally.
- Hydraulic system balancing of the heating surface and domestic hot water circulation systems to ensure correct volumetric flow rates.
- Installation of return temperature limiters (on the primary side) for indirect and direct house connections.
- In branched networks with several distributors on the secondary side, it is advisable to use a PICV upstream of the individual distributors for volumetric flow rate and differential pressure control.
- Existing pumps must be replaced by speed-controlled pumps if necessary (e.g. fault). In addition, the dimensioning must be checked in advance, as existing pumps are often over-dimensioned.

8.6.2 Heating system

- Application of the surface heating systems with correspondingly large heat transfer surfaces (e.g. floor or wall heating).
- Utilization of the temperature difference between the supply and return temperatures and reduction of volumetric flow rates.
- All heating surfaces must be equipped with automatic thermostatic valves consisting of actuator and control valve for room-by-room temperature control. The thermostatic valves must be provided with a pre-setting option.
- If no thermostatic valves are used (e.g. old building), suitable alternatives must be discussed with the customer or, if necessary, an optimisation strategy should be negotiated.
- Use of high return supply temperatures from individual heating circuits (e.g. ventilation, static heating surfaces, etc.) to supply other consumers (e.g. underfloor or wall heating).
- Use of heating circuit distributors with thermal separation between supply and return to avoid unregulated heat exchange.
- In order to guarantee a perfect function of the temperature control device, a hydraulic system balancing must be carried out.

8.6.3 Ventilation system

- Ventilation systems (individually or in exceptional cases in groups) shall be provided with control devices. Room temperature, supply air temperature and exhaust air temperature can serve as control variables. Straight-way valves can be used as control valves for direct connection. Both straight-way valves and three-way valves can be used for indirect connection.
- Ventilation systems fed with ambient air must be provided with an anti-freeze circuit and, if necessary, a start-up circuit.

8.6.4 Domestic hot water generation and circulation

- Avoid storage devices used solely for hot water with internal heat exchanger. Otherwise, a return supply temperature limitation or control must also be planned to avoid overheating.
- Domestic hot water heating with hard water (≥ 15 °fH; medium-hard) is possible. To prevent calcification of the heat exchangers, they must be connected with a return flow admixture (analog Figure 8.11 or Figure 8.12). With soft water or if the district heating supply temperature is always below 70 °C, an analogous connection is also possible as in Figure 8.8 (throttle circuit). Pump and non-return valve are not required.
- For hard water (≥ 15 °fH; medium hard), additional shut-off devices and flushing connections for cleaning the heat exchanger must be provided for large heat exchangers (> 200 kW). In the case of heat exchangers, in particular panel heat exchangers < 200 kW, a short-term replacement must be provided or kept in stock (stock-keeping by the operator).
- No exceeding of the prescribed hot water temperatures to prevent lime and salt build up in pipes, fittings and heat exchangers.
- For the heat exchanger, the connections for cold water and return supply temperature must be installed at the top for faster cooling.
- Reduction of unnecessary heat losses in hot-water circulation systems through thermal insulation of the pipes.
- If the hot water storage tank is dimensioned sufficiently, optimise the position of the temperature sensor and place it upwards. For larger hot water storage devices (≥ 500 liters), add a second temperature sensor.
- Temperature sensors must be installed above the register in hot-water storage devices with an internal heat exchanger.
- Water circulation must not occur in the hot area or in the vicinity of the temperature sensor.
- If several hot-water storage devices are integrated, connection in series is preferable to parallel connection. In low-load operation (e.g. mid-season) it should be possible to operate a single storage device. This reduces standby losses and improves hygiene.

8.6.5 Control engineering

- Use of tightly closing control elements and appropriate control to avoid uncontrolled convection.
- Avoidance of vibrations in control loops connected in series (e.g. control/ control of the customer installation) by faster reaction time and operation of the subsequent control loop.
- The hydraulic system and control engineering design must be carried out in accordance with the applicable technology. In particular, the requirements of Chapter 8.4 must be fulfilled:
  - Valve authority three-way valves ≥ 0,5
  - Valve authority straight-way valve ≥ 0,3
  - The circuits must be designed in such a way that the maximum permissible return supply temperature according to the technical connection requirements (TCR) can be maintained in every operation mode.

9 Economic efficiency

In order to assess the economic efficiency of district heating projects, the specific heat generation costs in centime per kWh, which are made up of the costs from the central heating plant and the heat distribution, must be determined. The heat production costs of possible decentralised heat generators in the supply area are also of interest as a basis for comparison.

Because the construction of heating systems and district heating networks is a long-term investment, an investment decision must take future development into consideration. For this purpose, a review is usually carried out from year to year over a period of, for instance, 20 years. For this period, an assessment of the risks in the event of deviations from the planned development is of great interest. Significant risks include failure to achieve the planned final capacity, reduced heating demands due to building refurbishment or changes in economic conditions, as well as changes in energy prices, the amount of investment and the interest rate on capital.

Although a temporal consideration is indispensable for an investment decision, an initial assessment is often made on the basis of the economic efficiency in the stationary situation. If this assessment is unattractive, a follow-up of the project in the planned form is usually unnecessary. Under certain circumstances, a comparison of alternatives can be made in advance on the basis of the stationary situation, for example by discarding uninteresting customers or partial networks. Depending on the question and timing of the project, different methods of assessment of profitability are therefore used, which are described in this chapter. However, before carrying out an economic analysis, the responsibilities must be defined, as explained in the introduction.

9.1 Responsibilities

The responsibility for assessing economic efficiency lies in general with the organizing institution of the project. The cost-efficiency calculation can be carried out by the client or commissioned by the client to the planner or a third body.

In any case, the task of the planner is to provide the required technical data and to advise the developers on the cost-efficiency calculation. However, the following basic economic assumptions for the economic analysis must be made by the developer:

- Expected capital interest rate (also expected rate of return).
- Calculated observation period or periods, if different observation periods are assumed for buildings and district heating network than for example those for the heat generator.
- Inflation rate (price increase).
- Increase in operating costs.
- Energy prices at project start, i.e. the electricity price and, depending on heat generation, prices for the main energy source(s) and additional energy sources.
- Energy price increases.
- Personnel costs (or, to simplify matters, a percentage of the investment).

It is advisable to agree on these basic assumptions in writing. The planning company provides the following technical data:

- Power and heat demand of the intended customers.
- Investment costs subdivided into basic components.
- Consumption-related costs, in the case of a fuel energy content of the fuel and fuel costs.
- Auxiliary energy consumption and resulting costs.
- Maintenance and restoration costs.

The detailed cost factors will be described in chapter 9.2. Responsibilities are usually divided between developer and planner as follows:

Duties of the developer:

- Determine which potential customers are to be taken into account.
- Determine the time of connection (influences the time of the investment costs incurred and the income to be expected from the sale of heat).

Duties of the planner:

- Determine the heat load demand including the load profile and the expected annual heat demand of the potential customers.
- Determine the investment costs for the connection of the potential heat customers.

9.2 Cost structure

According to VDI 2067 [117], the following four cost classes are distinguished when determining the costs of technical building installations:

- Capital-bound costs including repair and refurbishment,
- Consumption-based costs,
- Operational costs,
- Other costs.

In Table 9.1 the costs are allocated to the four cost groups. Individual cost types are estimated for the cost calculation based on benchmarks, for example as a percentage of the investment amount or the heat generated. It is necessary to define which cost types are assigned to the individual cost groups. According to VDI 2067, "repairs" includes all measures to maintain and restore the target condition with the three cost types "repair", "maintenance" and "inspection", which are defined as follows:

1. Repair comprises measures to restore the target state.
2. Maintenance includes measures to maintain the target state.
3. Inspection includes measures to establish and assess the current state.

Table 9.2 describes the cost types and shows which basic data is used for calculation. In some cases, indicative values are also given, which serve as a guide for heat generators with automatic wood heating systems. It should be noted that the specific costs depend on the fuel and the chosen technology and are additionally influenced by the size of the plant and the number of full-load operating hours.

Table 9.3 describes indicative values for the service life and repair costs in accordance with VDI 2067. The period of consideration to be taken for the cost-efficiency calculation shall be determined with the developer and, if necessary, with investors.

Table 9.1 Cost classes and cost types according to VDI 2067 [117].

Capital-bound costs	Consumption-bound costs	Operation-bound cost	Other Costs
Capital costs of plant components and buildings Repair costs	Fuel costs Costs for auxiliary energy Cost of operating materials Disposal costs Concession fees	Costs for operation, cleaning, maintenance and inspection Rent or lease Chimney sweep Emission measurement	Insurances Taxes Administrative expenses General contributions

Table 9.2 Cost types and basis for determining the annual costs and reference values. These values are exemplary for heat generators with automatic wood heating systems. Different to VDI 2067, the costs for maintenance are included in the personnel costs [21].

Cost type	Basis for determining the annual costs	Reference values for automatic wood heating systems
Capital costs of plant components and buildings (investments)	Investment sums of the subsection, Cost-accounting capital interest rate, Cost-accounting observation period	Useful life: Table 9.3 Capital interest rate according to country-specific situation and specifications of the builder-owner or investor
Repair costs (Repairs according to VDI 2067)	Investment sums of the subsection, Percentage sum of investment	
Fuel costs	Fuel consumption, calorific value and fuel price	effective quoted prices
Auxiliary energy (electricity) for heat generation and distribution	Percentage of heat quantity (generated or distributed) and electricity price	for heat generation: 1 % to 1.5 % p.a. of the generated heat, for heating network: 0.5 % to 1 % p.a. of the distributed heat
Operating materials, Heat generation	Price, consumption quantity	effective costs are to be estimated
Personnel costs without administration (operation, cleaning, maintenance, and inspection)	Effective personnel costs or simplifying Percentage of investment costs for heat generation	1.5 % p.a. of the investment costs of heat generation, but depending on the fuel (e.g. higher for waste wood)
Ash disposal	Fuel use, ash content, disposal method	effective costs are to be estimated
Concession fees, rent, lease, chimney sweep, emission measurement	Depending on the individual case	effective costs are to be estimated
Other costs	Percentage of the invested sum	0.5 % up to 1.5 % of the total investment

Table 9.3 Reference values for the operating life and the repair costs according to VDI 2067. The period of consideration to be assumed for the economic efficiency calculation is to be determined with the developer and possibly with investors. *The average operating life must be weighted with the planning costs for the individual subsections.

Subsection	Operating life in years	Specific repair costs in % of the annual investment costs
Plant components for heat generation (incl. regulation and control)	20	2 – 3
hydraulic system	20	2
Electrical and building services installation	20	2
Constructional facilities and development	50	1
Heat distribution network (incl. pipelines and earthworks)	40	1
Heat transfer stations	30	2
Vehicles	15	3
Planning	averaged*	0

9.3 Calculation of the heat production costs

For the analysis of energy systems, which usually have a long operational life and where future changes such as price increases are important, dynamic calculation methods for cost-efficiency analysis are useful. The changes expected during the period under consideration are estimated or given on the basis of the statistical averages in order to forecast the annual costs for the entire operational life of the plant.

Static procedures, on the other hand, only consider the conditions at the time of the realization of the plant. For periods of more than 15 years, there may be significant deviations from dynamic procedures. In many cases, dynamic methods can be based on simple assumptions, so that easy summation formulas can be used for cost-efficiency calculations.

The most important dynamic calculation methods are:

- the capital value method
- the annuity method
- the method of the internal interest rate.

For heat production costs, the annuity method is the usual calculation method and is described in detail in VDI 2067" [117]. The annuity method is used to determine the average annual costs incurred during the period under consideration. As a rule, this procedure is not sufficient to assess whether an installation can be operated economically. Further useful basics and hints for the application of the annuity method are provided in the documents of the impulse program RAVEL [60] and [61].

The VDI 2067 describes a calculation method that can be used if different rates of price increase are to be assumed for fuel costs and maintenance and repair costs. In the following, a uniform annual price increase e is accepted in a simplified way. In many cases, the following simplifications are possible as in the method described in detail in VDI 2067:

- The investments are only made at the beginning of the observation period.
- The observation period corresponds to the lifetime of the investments. This means that no replacement purchases need to be made within the observation period and that there are no residual values at the end.

Taking into account these simplifications, the **annual costs C_a** are calculated as follows:

$$C_a = C_I\ a + C_{OP}\ d_{NPV}\ a$$

C_I	Investment costs of the project
C_{OP}	Annual operating costs of the investment (fuel, maintenance and repair costs in relation to current prices)
a	Annuity factor: The annuity factor is calculated from the required rate of return i and the calculated observation period n.

$$\text{for } i = 0;\ a = \frac{1}{n} \qquad \text{for } i > 0;\ a = \frac{i\,(1+i)^n}{(1+i)^n - 1}$$

d_{NPV}	Present value factor (VDI 2067 [117]) or total discounting sum factor (RAVEL [61]). This is calculated from the required rate of return i, the price increase e and the calculative observation period n. Calculation according to VDI 2067 or corresponding tables.
d_{NPV} x a	Price dynamic annuity factor (VDI 2067) or average value factor (RAVEL).

Table 9.3 represents indicative values for the operational life and repair costs. The calculative observation period n can be determined on the basis of the operational life or specified by the developer, investor or a possible funding agency. If the observation period differs from the operational life, the following points must be taken into account, as stated in VDI 2067:

- For any capital expenditures during the observation period, their present value must be added to the initial investment I.
- If, at the end of the observation period, the end of the operational life has not yet been reached, investment I is reduced by the net present value of the residual value.

The annuity method provides the expected annual costs. The **heat production costs c_a** result from the reference to the usable heat Q_{UH} supplied to the customers:

$$c_a = \frac{C_a}{Q_{UH}} = \frac{C_I\ a + C_{OP}\ d_{NPV}\ a}{Q_{UH}}$$

In practice, the annuity method is often used in a simplified way without taking into account the price increase (inflation) and is used as a required rate of return, for example, the current interest rate for bank loans. With this simplification, the annuity method becomes a relatively inaccurate static consideration, which does not take into account the future development. When comparing two heating systems, this can lead to significant errors if the systems to be compared have different cost structures. However, in the context of a preliminary investigation, a system comparison with the current interest rate and without price increases is generally sufficient provided that the variants to be compared have similarly high proportions of the capital-linked cost types and a comparable price increase for the consumption and operational cost types.

If only the **heat production costs for the heat distribution network** (without heat generation) are to be considered, for example as a variant comparison of different grid structures, the following relations can be used. The costs are determined using the annuity method and are composed of the following components:

1. Capital costs
2. Operating costs:
   - Auxiliary energy costs especially electricity costs, amongst others for network pumps
   - Fuel costs to cover heat losses within the network
   - Costs for maintenance and operating.

As the costs shown here only cover the heat distribution, the costs for maintenance and upkeep are small compared to the other cost shares and are neglected in the calculation. The specific heat distribution costs thus result from the specific capital and operating costs:

$$c_{HD} = c_C + c_{OP}$$

The **specific capital costs c_C** result from the annual capital costs C_C in relation to the heat supplied to the heat consumers (usable heat):

$$c_C = \frac{C_C}{Q_{UH}} = \frac{\sum c_{PLS,i}\ L_{PLS,i}\ a}{Q_{UH}}$$

The **annual capital costs C_C** for the heat distribution network result from the sum of the specific investment costs for the individual partial lines multiplied by their route length and the annuity factor. The investment costs consist of the costs for material, laying, excavation work and restoration of the surface. For the specific investment costs per routed pipeline meter, for example, the guidance prices in in Table 13.8 for different pipe types can be used. A sub-section is defined as a route section with a constant nominal diameter of the supply and return pipeline from node to node and one node defines the branch of the line (house connection, network branching, etc.) or a change in the nominal diameter (extension, narrowing).

$$C_C = \sum c_{PLS,i}\ L_{PLS,i}\ a$$

The **specific operating costs c_{Op}**, consist of the specific pump costs, heat loss costs and maintenance and service costs:

$$c_{OP} = c_P + c_L + c_{maint.}$$

The **specific pump costs c_P**, result from the annual pump costs C_P. in relation to the heat supplied to the heat consumers (usable heat):

$$c_P = \frac{C_P}{Q_{UH}} = \frac{E_P\ P_{el}}{Q_{UH}}$$

The **annual pump costs C_P** are calculated from the energy demand of the network pumps (Chapter 7.4.5) and the electricity price:

$$C_P = E_P\ P_{el} = P_P\ \tau_{HC}\ P_{el}$$

The **specific heat loss costs c_L** result from the annual heat loss costs C_L in relation to the heat supplied to the heat consumers (usable heat):

$$c_L = \frac{C_L}{Q_{UH}} = \frac{Q_{L,a}\ c_{fuel}}{Q_{UH}}$$

The **annual heat loss costs C_L** cover the additional expenditure on fuel covering the heat losses in the network. The heat loss costs are calculated from the annual heat loss (Chapter 7.1.4) and the specific fuel costs:

$$C_L = Q_{L,a}\ c_{fuel}$$

The **specific fuel costs c_{fuel}** are calculated from the fuel price and the annual fuel use efficiency of the heat generator:

$$c_{fuel} = \frac{P_{fuel}}{\eta_a}$$

9.4 Comparison of different calculation options

The heat production costs described above can be determined to compare different heating systems over a longer period of time. It can be assumed that the inflation rate and the capital interest rate behave in a similar way and thus, if price increase remains constant, interest rates remain constant. With this assumption the calculation can be simplified using a so-called real interest rate.

The real interest rate corresponds to the interest rate that exceeds general inflation and is approximately the difference between the capital or nominal interest rate and the inflation rate. For example, with a nominal interest rate (capital interest rate) of 5% p.a. and a general price increase of 3% p.a., the real interest rate is 2% p.a. In the economic efficiency calculation, interest rate and inflation rate may not be determined independently, as they are closely linked. A distinction must therefore be made between nominal and real perception.

In nominal perception (VDI 2067), the price increase of the later years is taken into account for the determination of costs and yields. In this case, the required rate of return shall be the interest rate to be paid effectively on bank loans.

In the simplified real perception, current prices are expected over the entire observation period. In this case, the required rate of return shall be the real interest rate (difference between the interest rate and the rate of inflation). It should be noted that although the ratio of costs is correctly reflected, the calculated heat production costs do not correspond to the actual ones. For a simple calculation, it is also assumed that the repayment of the installments will take place at the same prices in real terms as originally after the end of the operational life. It should be kept in mind that the operational lives of different systems may vary considerably. The observation period considered must then correspond to the longest average operational life of the different systems. Further information can be found in the documentation RAVEL [61].

If the above rules are followed, the real and nominal perspectives lead to the same qualitative results. If price increase is not taken into account when comparing the heat production costs of different heating systems, the real interest rate must be used as the required rate of return.

In order to take environmental aspects into account when comparing energy systems, the external costs of environmental damage can be considered as energy price increases. Table 9.4 presents the energy price surcharges proposed by the Confederation in Switzerland in the reference year 1997. Part of the external costs is now covered by the CO2 tax on fossil fuels. As of 1 January 2016, the CO2 tax in Switzerland was CHF 84 per ton of fossil CO2. For heating oil with a calorific value of around 10 kWh per liter and CO2 emissions of around 2.65 kg per liter, this corresponds to approximately 2.24 centimes per kWh or half of the imputed surcharge proposed for heating oil in 1997.

Table 9.4 Energy price surcharges to reflect external costs in Switzerland (based on Swiss Federal Office of Energy and Federal Office for the Environment; Situation 1997) [21].

Energy Carrier	Example Energy Price	Energy Price Surcharge	Example Energy price surcharge with external costs
	CHF/MWh	CHF/MWh	CHF/MWh
Fuel	55	45	100
Fuel gas	60	30	90
Wood	40	15	55
Electricity	150	50	200

9.5 Business plan, budgeted balance sheet and budgeted income statement

The term "business plan" comes from the USA and describes a business concept or a company plan and thus a business idea that is to be implemented in a company. A business plan is thus also an instrument that provides information about the quality of a company. For the following, the business plan comprises a document in two parts:

1. Initial Text Array: This is the business plan as a written formulation of the business idea with regard to product, service, customers and marketing. The business plan provides information about the development of the company and enables an assessment of the risks. For a district heating network, it is important to show the effects of different kinds of network expansion.

2. Financial section: The budgeted balance sheet and the budgeted income statement for planned income and expenditure, subsidies, financing and liquidity planning. The budgeted balance sheet and budgeted income statement are generally prepared over a period of at least 20 years.

In these two parts, the business plan should cover the following elements:

- Executive Summary (maximum two pages)
- Company (founding team, company profile, company goals)
- Product or service: customer advantages/benefits, status of development, production
- Sector and market: sector analysis, market analysis and market segmentation, target customers, competition, location analysis
- Marketing: Entering the market, marketing and sales concept, sales promotion
- Management and key persons
- Implementation planning
- Opportunities and risks
- Financial section: Planning for the next 20 to 25 years: personnel planning, investment and depreciation planning, budgeted profit and loss account, liquidity planning, listing of financial requirements.
- Sensitivity analysis as a supplement to investment calculations, in which the following questions are answered by entering the most important input variables:
  - Which input variables have a particularly strong influence on the result variable?
  - Within which limits can the values of the input variables fluctuate without endangering the success of the company?
- Important input variables include the following:
  - High proportion of borrowed funds in financing and the interest rate on borrowed capital
  - Fuel price (and guarantee of supply)
  - Exceeding construction and plant costs
  - Electricity price
  - Personnel costs
  - Unsecured subsidies.

A business plan should be written by the developer. Not only is the developer responsible for the business idea but is also interested in representing the business plan to the outside world. The task of the planner is to support the developer in the preparation of the business plan.

The following sections discuss the **budgeted balance sheet and the budgeted income statement**.

The assessment of the economic efficiency of an enterprise cannot be based solely on the calculation of the average heat production costs. Even if the heat production costs are lower than the revenues over a longer period of consideration, it is not possible to compensate for any losses in the first years of operation by profits in later years if the liquidity is not secured for this purpose. Special attention must therefore be paid to the liquidity situation. The purpose of the budgeted balance sheet and budgeted income statement is to assess the economic situation over each year.

The division of tasks for this must be defined at the start of the project. Framework conditions such as the structure and linear heat density of the heating network influence the cost-efficiency, therefore it makes sense for the planner to participate in the economic optimisation process, as it is the planner who collects the relevant data.

However, the cost-efficiency assessment and the associated planning calculations should not only be carried out during the planning phase, but should be regularly reviewed during the project period and supplemented by considering possible cost optimisations.

When calculating the budgeted balance sheet and the budgeted income statement for the individual years, it must also be clarified whether and how price increase is to be taken into account. However, the simplified real approach (see chapter 9.4) is not permissible, because the difference between the real interest rate and the bank interest rate increases with a high inflation rate. This leads to a correspondingly higher interest burden in the first few years. Using the real interest rate can therefore be an underestimation of the cost of capital with a corresponding risk, especially in the first few years.

If price increase is not taken into account at all (nominal interest rate and no price increases), the interest burden in the planned income will be slightly higher. With low inflation, this deviation can be neglected. If the forecast income is used as a tool for optimizing the heating network, inflation can also be neglected. Price increase affects both expenditure and revenue and is generally negligible compared to other uncertainties.

The budgeted balance sheet and budgeted income statement are usually part of the documents (usually the business plan) submitted to a potential creditor. The assumptions regarding interest and price increase should be discussed and agreed upon with the creditor.

Optimisation

10 Analysis and optimisation

10.1 Method

The analysis of the installations at the customer is based on the fact that the temperature difference between the supply and return determines the transferable heat output of the district heating network. This temperature difference is strongly influenced by the installations. If installations at the individual customers do not cool the district heating water sufficiently, this increases the return temperature and reduces the capacity of the network. At the same time, the energy consumption for pumps and the heat losses of the system increase as the mass flow rate and the system temperature go up.

As a methodological approach, the quality of the customers installations and the optimisation potential for the system are determined by examining the amount of heat and water transferred. This approach has been used by a Swedish planning office on numerous district heating networks [12] and has also been used in Switzerland [13]. A practical survey on two district heating networks [17] identified that one customer installation in each significantly reduced the efficiency and capacity of the network. With optimisation measures implemented on only this one substation, the entire primary return temperature could be reduced by 1.5 K or 1.2 K. A cost-efficiency calculation showed that the costs for optimisation are amortized in 2.3 and 3.9 years respectively and there is a higher return on investment. Additionaly, the optimisation resulted in an increase in capacity or a reduction in the volumetric flow rate of 5% and 4% respectively.

In order to determine which substations have the greatest influence on the overall primary return temperature in the network, the quantity of additional water in existing district heating networks (**excess consumption**) is determined. This quantity is compared to operation at reference temperature difference.

This calculated influence on the return temperature determines the temperature difference by which the total primary return of the network decreases. Although large customers are generally more important for the cost-effectiveness of a district heating network than small ones, all customers must always be taken into account when analysing the quality of the system. If the reference temperature difference is significantly undercut, even a single small substation can influence the entire network.

10.1.1 Basics

The following equation for the heat flow (heat demand per capacity) forms the basis for the calculation of the excess consumption and the influence on the return temperature.

If the density and the specific heat capacity of the water are assumed to be constant in the temperature range under consideration, the transferable heat output is proportional to the temperature difference between supply and return flow and the following applies:

$$\dot{Q} = \dot{m}\, c_p\, \Delta T = \dot{V}\, \rho\, c_p\, \Delta T$$

To evaluate customers, the heat meter data according to Figure 10.1, as well as heat and water quantities over an observation period are considered and the average temperature difference is calculated.

Ideally, data is collected continuously with an evaluation over a period of, for example, one month and one year, as well as an optional daily evaluation for monitoring purposes. If the data is collected manually, an evaluation over at least one quarter in the heating period is recommended.

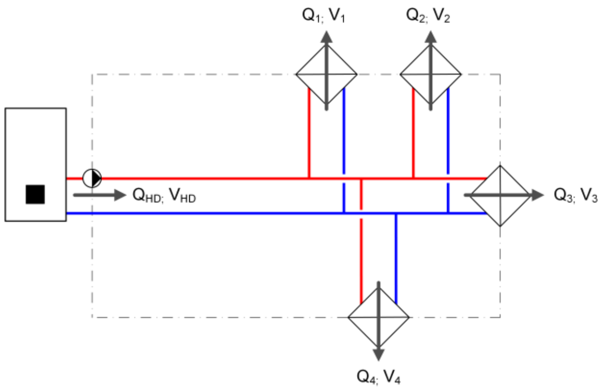


Figure 10.1 Representation of the characteristic values for recording heat meter data for a district heating network with a heat generator with main group for heat distribution HD and four customers V1 to V4.

10.1.2 Excess consumption

The excess consumption is the additional volume of water that has flowed over the transfer station in the considered period in comparison to the volume at a reference temperature difference. The excess consumption for the customer i is calculated by measuring the difference between the measured volume and the ideal volume (related to a reference temperature difference) as follows:

$$\underset{\text{Excess Consumtion}}{\Delta V_{i,EC}} = \underset{\text{Heat Meter Vol.}}{\Delta V_{i,HM}} - \underset{\text{Reference Vol.}}{\Delta V_{i,REF}}$$

The measured volume difference for the customer i in the observation period, is the difference between the heat meter reading at the end time (t1) minus the heat meter reading at the start time (t0) as follows:

$$\Delta V_{i,HM} = V_{i,HM}(t_1) - V_{i,HM}(t_0)$$

The ideal volume for the customer i in the observation period at the reference temperature difference is calculated by the ratio of the measured heat quantity difference for the customer i in the observation period to the density of the water, the specific heat capacity of the water and the reference temperature difference as follows:

$$V_{i,REF} = \frac{\Delta Q_{i,HM}}{\rho\, c_p\, \Delta T_{REF}}$$

The heat quantity difference for the customer i in the observation period is calculated by measuring the difference between the heat meter reading at the end time (t1) minus the heat meter reading at the start time (t0) as follows:

$$\Delta Q_{i,HM} = Q_{i,HM}(t_1) - Q_{i,HM}(t_0)$$

The reference temperature difference should be based on the maximum of the temperature data of the supply and return of the technical connection requirements (TCR). If no TCR exist, a technically feasible value should be chosen as a reference.

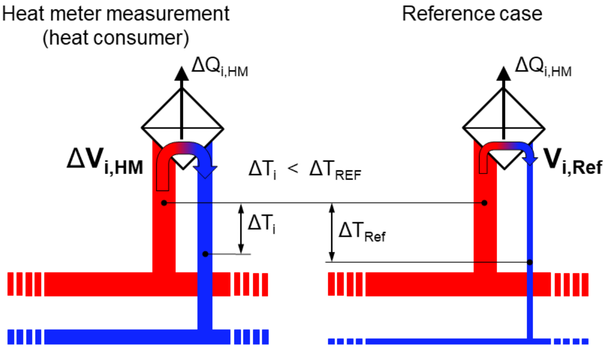


Figure 10.2 Representation of the excess consumption for a customer by comparing the water volumes from the heat meter data and with a reference temperature difference.

10.1.3 Impact on primary return temperature

A mere assessment of the excess consumption is not sufficient to assess the need for optimisation. Therefore, the influence that the optimisation of one transfer station (based on the reference temperature difference) has on the temperature of the entire primary return flow is of particular interest. The influence on the return temperature for each customer installation is represented by a temperature difference which describes how much the temperature of the entire primary return decreases when this installation is optimised for the reference temperature difference (Figure 10.3).

The value is calculated by subtracting the average temperature difference from the heat meter data of the main meter and the average temperature difference with an optimised transfer station as follows:

$$\Delta T_{i,RTN} = \Delta T_{AS} - \Delta T^*_{AS}$$

The average temperature difference from the heat meter data of the main meter is the ratio of the heat quantity difference for the main meter in proportion to the density of the water, the specific heat capacity of the water and the volume difference for the main meter in the observed period as follows:

$$\Delta T_{AS} = \frac{\Delta Q_{HD,HM}}{\rho \, c_p \, \Delta V_{HD,HM}}$$

The average temperature difference with an optimised transfer station is the ratio of the heat quantity difference for the main meter in proportion to the density of the water, the specific heat capacity of the water and the difference of the volume difference for the main meter in the observation period minus the excess consumption for the individual customer as follows:

$$\Delta T^*_{AS} = \frac{\Delta Q_{HD,HM}}{\rho \, c_p \, (\Delta V_{HD,HM} - V_{i,EC})}$$

The heat quantity difference for the main meter, in the observation period, is calculated from the difference between the heat meter reading at the end time (t1) minus the heat meter reading at the start time (t0) as follows:

$$\Delta Q_{HD,HM} = Q_{HD,HM}(t_1) - Q_{HD,HM}(t_0)$$

The volume difference for the main meter, in the observation period, is calculated by the difference between the heat meter reading at the end time (t1), minus the heat meter reading at the start time (t0) as follows:

$$\Delta V_{HD,HM} = V_{HD,HM}(t_1) - V_{HD,HM}(t_0)$$

In summary, the influence on the return temperature can be calculated in the following equation:

$$\Delta T_{i,RTN} = \frac{\Delta Q_{HD,HM}}{\rho \, c_p} \left(\frac{1}{\Delta V_{HD,HM}} - \frac{1}{(\Delta V_{HD,HM} - V_{i,EC})} \right)$$

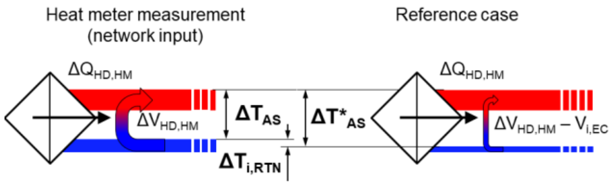


Figure 10.3 Representation of the temperature difference and the volumetric flow rates before optimisation (left) and after optimisation (right) of an individual customer installation.

10.2 Procedure

10.2.1 Data collection and analysis

The reading of the heat meters for the calculation of the delivered heat usually takes place once to four times a year, whereby usually only the heat quantities are recorded.

Data collection: The easiest way to record data is to collect heat meter data in a central location. This can be done, for example, via a data server that collects the consumers heat meter data. Even more convenient is a higher-level control system that collects all relevant data. This facilitates evaluation and invoicing. For the evaluation of the heat meter data, in addition to the heat quantity in kWh, the volume in m^3 must also be recorded. If no control system or similar is available, the following options exist: monthly reading of the heat meter data on site (extensive), quarterly reading or seasonal reading of the heat meter data (heating season, transition period, summer) or reading at the beginning and end of the accounting period.

Evaluation: The easiest way to calculate excess consumption and determine the influence on the return temperature is by creating a spreadsheet. The following explanations are based on an exemplary data evaluation in an Excel spreadsheet using a fictitious district heating network.

In the first part, the table contains an **input area** (Figure 10.4), in which input sizes (**red**) are to be set. In the example shown, the following values are entered:

- Reference temperature difference ΔT_{REF} 35 K
- Specific heat capacity water c_P 4.185 J/(kg K)
- Density water ρ_W 980 kg/ m³
- Heat meter reading period Start Jan. 31 2016
- Heat meter reading period End May 30 2016

The second part includes an output area (Figure 10.4) for **calculated variables** (**blue**) in the meter reading period, which are determined by the input variables and the heat meter data. The following applies in the example:

- Meter reading period t in days 120 days
- Meter reading period t hours 2880 h
- Mean temperature difference 28.2 K
- Total heat quantity 4’751’446 kWh
- Total water volume 147’885 m³
- Specific water quantity 51 m³/h

In a third input area, **prepared heat meter data** is inserted (Figure 10.5). The excess consumption and the influence on the return supply temperature are determined from this data and the input variables. The five columns with red background are reserved for the the consumers prepared heat meter data. The heat meter data must be prepared in a separate document and then copied into the document. The preparation takes place according to the following criteria:

- Customer number: e.g. meter number of the heat meter
- Customer name
- Subscribed capacity: in kW (not absolutely necessary, here to forecast the annual number of full operating hours of the customers)
- Heat quantity: transferred heat quantity in kWh within the reading period
- Quantity of water: flowed through quantity of water in m³ within the reading period.

The following values for each heat customers are calculated in the following columns:

- **Full-load operating hours**: Calculates the annual full-load operating hours of the customers. The value is extrapolated to one year if the reading period is shorter than one year (8760 hours). This calculation step is not necessary for the evaluation and serves for the plausibility check of the heat meter data.
- **Excess consumption**: Presents the results of the over-consumption related to chapter 10.1.2.
- **Influence on return flow**: Shows the results for the influence on the return temperature according to chapter 10.1.3. It should be noted that the calculated values of the influence on the return temperature are usually not reached during an optimisation, but they do show the optimisation potential.
- **Average temperature difference**: The average temperature difference is calculated and displayed for the recorded meter reading period. This information is helpful to additionally assess the results of the additional consumption and the influence on the return temperature.

With the Excel function Filter and Data Ascending Sorting, the customers can be ordered in the front column 'Rank' with descending excess consumption. The customer with the highest excess consumption is ranked 1st.

Assumptions		
Reference Temperature difference	K	35
Specific heat capacity	kJ/(kg K)	4.185
Water density	kg/m3	980
Meter reading period	31.01.2016	
	30.05.2016	
	d	120
	h	2880
Outcomes		
Mean temperature difference	K	28.2
Total heat quantity	kWh/t	4'751'446
Total volume of water	m3/t	147'885
Specific water volume	m3/h	51

Figure 10.4 Excel spread sheet for the evaluation of excess consumption and influence of the return temperature for a fictitious district heating network. The column presents input variables and calculated variables.

Rank	Consumer number	Name of heat consumer	subscribed heat load	ΔHeat quantity	ΔWater volume	Full-load operating hours	Excess consumption	Influence on return flow	Mean temperature difference
–	–	–	kW	kWh/t	m3/t	h/t	m3/t	°C	K
1	WZ_27	Wärmeabnehmer 27	980	638'550	42'943	1'982	26'928.70	-6.3	13.1
2	WZ_60	Wärmeabnehmer 60	550	286'696	12'562	1'586	5'371.91	-1.1	20.0
3	WZ_10	Wärmeabnehmer 10	500	436'239	14'030	2'654	3'089.49	-0.6	27.3
4	WZ_48	Wärmeabnehmer 48	830	876'830	24'859	3'213	2'868.84	-0.6	31.0
5	WZ_40	Wärmeabnehmer 40	60	47'055	3'456	2'385	2'275.90	-0.4	12.0
6	WZ_23	Wärmeabnehmer 23	48	48'818	2'442	3'094	1'217.69	-0.2	17.5
7	WZ_67	Wärmeabnehmer 67	60	69'977	2'526	3'547	771.04	-0.1	24.3
8	WZ_17	Wärmeabnehmer 17	10	10'415	849	3'168	587.80	-0.1	10.8
9	WZ_59	Wärmeabnehmer 59	50	47'507	1'754	2'890	562.56	-0.1	23.8
10	WZ_14	Wärmeabnehmer 14	20	13'018	610	1'980	283.52	-0.1	18.7

Figure 10.5 Excel spreadsheet for evaluation of excess consumption and the influence of return temperature. Shown are the ten worst customer installations of a fictitious district heating network.

10.2.2 Evaluation

On the basis of the data analysis, the worst heat integration at the customer with the highest excess consumption can be determined. The example shows that the customer 27 has the highest excess consumption and contributes to the high return temperature with an average temperature difference of only 13.1 K. The optimisation of the heat integration at the customer could cool the primary return temperature by 6.3 K.

The next three end users together have an optimisation potential to lower the primary return temperature of just over 2.3 K. In comparison, the other customers have only a small influence on the return temperature.

If an uncharacteristically small value appears in the column 'Mean temperature difference', this indicates a possible malfunction in the transfer station. This should be fixed as soon as possible.

The next step is to find out the reasons for the high excess consumption of the worst heat integration at the end user and to develop and implement suggestions for improvement. For this purpose, it is recommended to visit the customer with the operator of the district heating network and a specialist. Optimisation measures can be derived from this, which are usually to be carried out by a specialist in district heating technology. Before making a decision, the economic effect of the individual optimisation measures can be assessed in advance if necessary.

Typical reasons for insufficient temperature difference and subsequent excess consumption are badly designed or improperly installed components, wear and tear due as well as inappropriate settings of the measurement and control technology. According to an analysis by the International Energy Agency (IEA), three categories of plant technology are responsible for the defects [12]:

- The secondary heat distribution is responsible for about 60% of the disturbances.
- Domestic hot water generation accounts for 30 % of the damages.
- The remaining defects are the responsibility of the transfer station with primary and secondary components.

Causes of malfunctions include defective valve controls, leaky valves, inappropriate controller settings and inappropriate hydraulic integration on the primary and secondary sides of the customer installation. Examples are open expansion vessels, double distributors, bypasses, overflow devices between supply and return flow, injection circuits with three-way valves, deflection circuits with three-way valves and four-way mixers. Due to the numerous possible sources of error, a survey of 52 district heating networks shows that many networks do not reach the intended temperature difference in practice [16].

10.2.3 Implementation and performance analysis

The operator of the district heating network is responsible for optimisation and also benefits from them. The implementation of optimisation measures can therefore usually be carried out by the network operator under sole responsibility.

In contrast, there is a possible conflict of objectives for measures on the secondary side, since the increased temperature difference primarily benefits the heat supplier, whereas the responsibility for implementing these measures lies with the end user. Although the end user benefits from reduced pump capacity and reduced distribution losses on the secondary side, the resulting cost savings are usually only minor. For this reason, it is necessary to examine whether minimum requirements for temperature difference and measures are to be derived from the TCR and/or whether a contribution to the costs for the heat supplier is possible.

After the measures have been implemented, the optimisation should be checked with a performance analysis.

10.3 Recommendations for the procedure of customer installation analysis

The analysis of the installations at the end user on the basis of the excess consumption is easy to implement and the basis for this is mainly heat meter data of the customers and heat generators. Basic knowledge in thermodynamics and practical knowledge or experience in building and district heating technology is required. Depending on available technologies and know-how, the procedures presented in Table 10.1 can be used by operators of district heating networks themselves or by an experienced specialist (planner).

Information on analysis of the installations at the end user and the implementation of measures:

- It is recommended to **collect all heat meter data** (end users and heat generators) in one central location. This can be done, for example, via a data server that records the consumers´ heat meter data. Even more convenient is a higher-level control system that records all relevant data. This facilitates evaluation and invoicing. For the evaluation of the heat meter data the volume in m3 must be recorded during the observation period in addition to the usual heat quantity in kWh.
- The **evaluation of the heat meter data** on the basis of the excess consumption is usually carried out using a spreadsheet (e.g. Excel). A certain initial effort is required for the analysis. If the heat meter data is presented in an easily understandable way, the time expenditure is relatively small.
- It is recommended to **periodically analyze** all installations at the customer for excess consumption. This way changes e.g. by defect valve actuators can be identified quickly. Depending on the complexity and degree of automation of the evaluation, this should be done at least per quarter, preferably monthly. With a control system it is possible to automate this evaluation and to report the additional consumption, e.g. on a monthly basis.
- In order to identify the need for optimisation, it is necessary for **experienced personnel** or **external specialists** to clarify the situation on the site. Together with the end user, the present condition must be analysed in order to derive optimisation measures. A ranking of the possible optimisation measures according to use and costs is recommended. This facilitates the decision makers to find the best possible solution.
- When implementing optimisation measures, a distinction must be made between **the area of responsibility** as well as the **area of authority**. On the one hand, the optimisation measure affects the primary side; in this case the responsibility lies with the heat supplier. On the other hand, optimisation can also affect the secondary side, in which case the responsibility lies with the respective customer. In both cases, the optimisation measure mainly benefits the heat supplier by increasing connection capacities and lowering return temperatures and, as a result, lowering heat production costs. If the optimisation measure concerns the secondary side, the heat supplier must estimate the extent to which cost sharing is possible. It should be contractually agreed between the customer and the heat supplier that, in the event of a secondary refurbishment of the hydraulic integration, all measures must be approved by the heat supplier.
- Once the measures have been implemented, a **performance analysis** should be carried out to assess and confirm the optimisation results.

Table 10.1 Procedure for the analysis and optimisation of customers.

What?	Description	Who
Data acquisition and evaluation	Recording heat meter data	**Operator** / Planner
	Identifying excess consumption and influence on return supply temperature	Operator / **Planner**
	Ranking of customers according to excess consumption	Operator / **Planner**
Validation	On the basis of data acquisition and evaluation, the worst heat integration at the customer is determined for an in-depth assessment.	**Operator / Planer**
	Assessment of the actual situation of the worst heat integration at the customer on site	**Operator / Planner / Customer**
	Definition of optimisation measures	Operator/ **Planner**
	Weighting of optimisation measures according to benefit and effort	Operator / **Planner**
Implementation and performance analysis	Clarify responsibilities	**Operator** / Customer
	Weighting of the costs or cost sharing if the optimisation measure concerns the secondary side (responsibility on customer end)	**Operator**
	Implementation of optimisation measures	**Operator / Customer**
	Analysis of optimisation	Operator / **Planner**

Appendix

11 Material properties of water

Table 11.1 Material properties of water within saturation state [48].

Temperature °C	Steam Pressure bar	Density kg/m³	Specific Heat Capacity kJ/(kg K)	Heat Conductivity W/(m K)	Kinematic Viscosity 10^{-6} m²/s
0	0.0061	999.79	4'220	0.562	1.792
0.01	0.0061	999.79	4'220	0.562	1.792
5	0.0087	999.92	4'205	0.572	1.518
10	0.0123	999.65	4'196	0.582	1.306
15	0.0171	999.05	4'189	0.591	1.139
20	0.0234	998.16	4'185	0.600	1.003
25	0.0317	997.00	4'182	0.608	0.893
30	0.0425	995.61	4'180	0.615	0.801
35	0.0563	994.00	4'179	0.622	0.724
40	0.0738	992.18	4'179	0.629	0.658
45	0.0959	990.18	4'179	0.635	0.602
50	0.1235	988.01	4'180	0.641	0.554
55	0.1576	985.67	4'181	0.646	0.511
60	0.1995	983.18	4'183	0.651	0.474
65	0.2504	980.53	4'185	0.655	0.442
70	0.3120	977.75	4'188	0.660	0.413
75	0.3860	974.83	4'192	0.663	0.388
80	0.4742	971.78	4'196	0.667	0.365
85	0.5787	968.60	4'200	0.670	0.344
90	0.7018	965.30	4'205	0.673	0.326
95	0.8461	961.89	4'211	0.676	0.309
100	1.0142	958.35	4'217	0.678	0.294
110	1.4338	950.95	4'230	0.681	0.268
120	1.9867	943.11	4'246	0.684	0.246
130	2.7026	934.83	4'265	0.685	0.228
140	3.6150	926.13	4'286	0.685	0.212
150	4.7610	917.01	4'310	0.684	0.199
160	6.1814	907.45	4'338	0.682	0.188
170	7.9205	897.45	4'369	0.679	0.178
180	10.0260	887.01	4'406	0.675	0.169
190	12.5500	876.08	4'447	0.670	0.162
200	15.5470	864.67	4'494	0.663	0.155

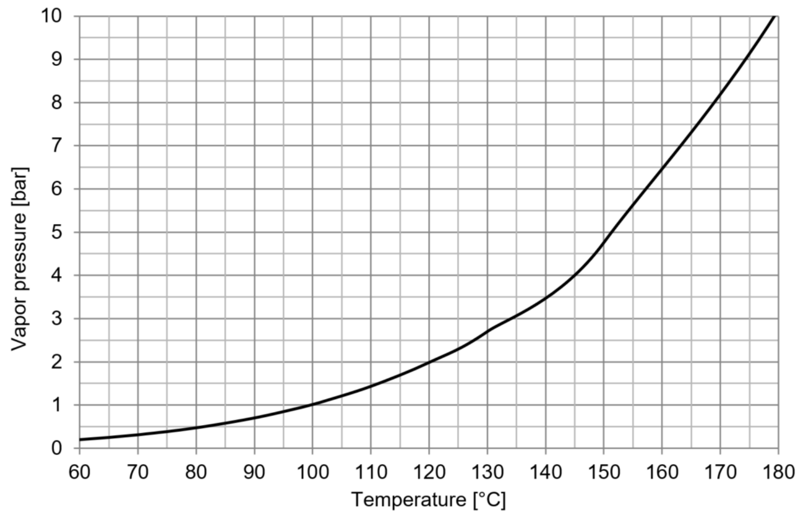


Figure 11.1 Vapor pressure of water in the saturation state [48] referred to Table 11.1.

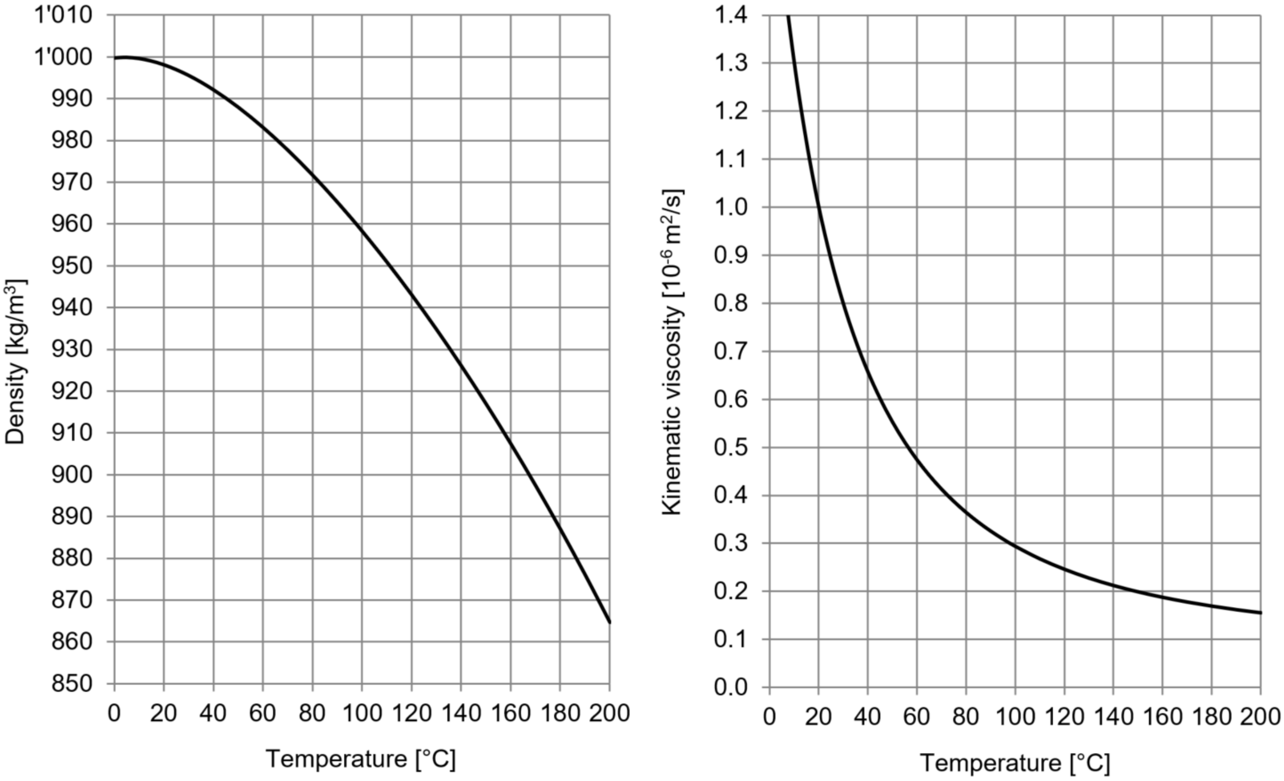


Figure 11.2 Material values of water in saturation state [48] referred to Table 11.1.
On the left density, on the right kinematic viscosity

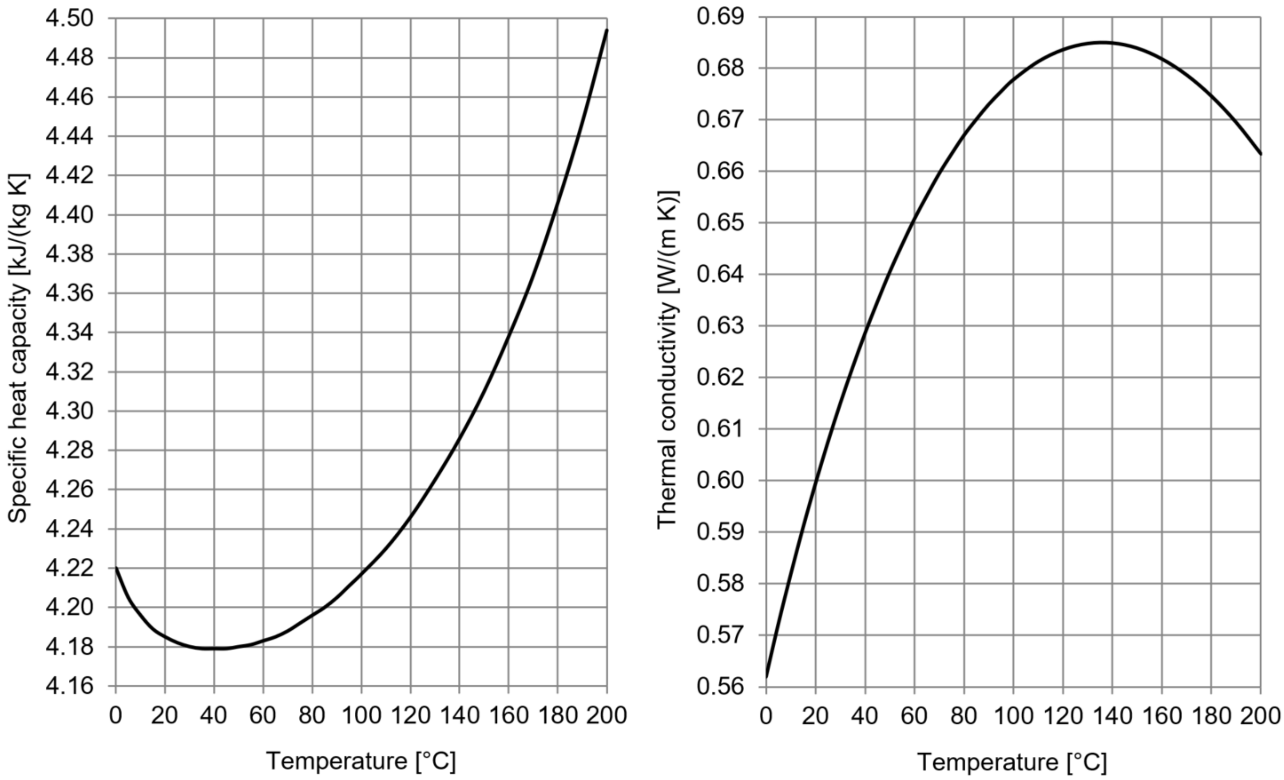


Figure 11.3 Material values of water in saturation state [48] referred to Table 11.1.
On the left specific heat capacity, on the right heat conductivity

12 Supplements for water quality

The following explanations are intended as a supplement to chapter 4.8 Water quality.

12.1 Measured values

The quality of circuit water within a district heating network can be guaranteed by metering and monitoring the following values:

pH-Values:
The pH-value represents a dimension value that represents the acidic, neutral or alkaline reaction of the water. Acids separate hydrogen Ions (H+). Lyes (bases) separate hydroxide ions (OH–). Therefore, the pH-Value is the measurement for hydrogen ion concentration (H+).

$$pH = -\log_{10}(H+)$$

Example: 0.00001 mol H+/l: pH = $-\log_{10}$ 0.00001 = 5

The pH value scale ranges from 0 to 14 at a temperature of 25 °C, water is acid for pH < 7, neutral at pH = 7 and basic at pH > 7. A change to the pH value by one point (for example pH 8 to pH 9) means that the acidity of water increases by ten.

Measuring is conducted using indicators (indicator paper, liquid indicators) or electrochemically by using pH-Electrodes.

Electric Conductivity:
Determining salinity is generally done by measuring electric conductivity. All dissociated constituents of the medium are examined (e.g. bases, acids and salts). Increased conductivity stimulates electrochemical corrosion.

Electric conductivity represents an inverse value of the electrical resistance (S = 1/Ohm) with respect toto a water cube of 1 cm at 25 °C (Unit: µS/cm; Micro Siemens/cm).

Water Hardness:
The SI – measurement system measures the concentration of the alkaline earth ions or in other words the water hardness as mol per liter or for lower concentrations as millimoles per liter (mmol/l).

In former times water hardness was defined using the German hardness grade classification (°dH). A single German hardness grade equals 10 mg calcium oxide per litre or 10 g calcium oxide per cubic meter. The French hardness grades (°fH) were common within Switzerland. Later the specifications of water hardness changed to the more practical unit milliequivalent per liter (meq/l) as represented in Table 12.1. Nowadays the molar classification is required by law.

Table 12.1 Translation of water hardness.

		°dH	°fH	meq/l	mmol/l
German Grades	**°dH**	1	1.78	0.357	0.1783
French Grades	**°fH**	0.56	1	0.2	0.1
Equivalent Unit	**meq/l**	2.8	5	1	0.5
SI-System of Units	**mmol/l**	5.6	10	2	1

Regional water quality can differ enormously. Water hardness is represented by the dissolved concentrations of calcium and magnesium salt (total hardness TH). In addition to this there is a distinction between carbonate hardness (CH) and non-carbonate hardness (NCH). Both hardening constituents are crucial due to the fact that over 61°C deposits occur (boiler scaling) in pipes and on vessel walls.

- TH: Sum of the alkaline earth TH:
  TH = calcium Ca^{2+} + magnesium Mg^{2+}
- CH: The proportion of the share of calcium and magnesium, that exists as hydrogen carbonate (HCO_3^-) and carbonate (CO_3^{2-}).
- NCH: The proportion of calcium and magnesium that are connected to other anions such as, chlorides (Cl^-), sulfates (SO_4^{2-}), nitrates (NO_3^-)

The following rule applies: **TH = CH + NCH**

In Switzerland water hardness is divided into six hardness grades (see Table 12.2). Water quality varies regionally. General regional specifications are insufficient. Water values of interconnected systems using salt water, ground water and spring water vary during the course of the year.

Table 12.2 Swiss water types and the classification within French Hardness grades

Water Type	Water Hardness °fH
soft	0-7
moderately soft	7-15
slightly hard	15-25
rather hard	25-32
hard	32-42
very hard	> 42

It is recommended to ask the responsible water supplier to carry out an anlysis. Drinking water is considered to be a food and must be classified by the water supplier. If surface and rain water are used, a water analysis must be be provided as well.

Oxygen
Normal tap water consists of 5 – 12 mg/l oxygen depending on the source. After a few days or weeks lower oxygen content will result within a closed water circuit if no further feeding occurs.

An increasing oxygen content within a closed system (heating or cooling circuit) is unwanted and can be the result of the introduction of make-up water or oxygen diffusion that results from the pipe system (e.g. by permeable plastic pipes).

For the determination of the oxygen content of circuit water it must be ensured that no additional oxygen can access the sample water.

Chloride
The chloride content is essential factor for corrosion. Chloride enhances local corrosion (pitting, shallow pit formation). Increased chloride concentrations can also affect chromium steel and chromium-nickel steel especially at high wall temperatures. The chloride level is measured in mg/l.

Sulphate
Sulphates are also crucial factors for corrosion because they affect local corrosion (pitting, shallow pit formation). The sulphate level is measured in mg/l.

Nitrates
Nitrates are also responsible for local corrosion (pitting, shallow pit formation). High nitrate concentrations in drinking water especially in areas of intensive agricultural utilization can affect the corrosion behavior. The nitrate level is measured in mg/l.

12.2 Methods of water treatment

Methods of water treatment can depend on make-up water and/or to a partial flow of the circuit water. The water quality can be decreased by the inflow of outside water, gas intrusion or corrosion processes and can be altered by the use of conditioning agents. With the help of a partial flow treatment plant (filtration, decarburization and demineralization) the suspended and dissolved substances can be removed using a bypass. The following procedures are possible:

Filtration:
In order to remove water-insoluble agents and prevent sedimentation or malfunction of downstream components, various mechanic procedures can be applied. Tube and bag filters and precoat cartridge filters are used in order to remove highly dispersed agents. Dirt traps are utilized for coarse particles.

Demineralization:
Dissolved salts (cations and anions) can be removed by using the ion exchange technique in order to lower the electrical conductivity. Extremely acidic cation exchangers are combined with extremely basic anion exchangers. A higher demineralization rate can be reached by the use of membrane technology (reverse osmosis). Temperature specifications have to be considered while utilizing these methods during partial flow operation.

The electrode ionization (EDI) is an electro-chemical demineralization method. The procedure is a combination of electro dialysis and demineralizer. An EDI-device is used for final demineralization following reversed osmosis. A conductivity rate of 0.06 µS/cm can be reached under optimal pre-treatment. An EDI-device consists of ion selective membranes and ion exchange resins that form a layer between two electrodes (anode (+) and cathode (-)). By adding direct current (DC) voltage to the poles ions move according to their electric charge. Only anions are permitted to move through anion selective membranes. Cations are allowed to pass through cation selective membranes only. Both types of membranes are impervious to water.

Softening:
By using regenerative cation exchangers that operate with technical halite, the hardening constituents (calcium and magnesium ions) are replaced by sodium ions. The water is descaled and cannot cause anymore scale formation. This method of water purification is useful in a saline operating mode.

Deaeration:
Thermal deaeration and vacuum degassing (pressure stages degassing) are standard procedures to remove the natural portion of dissolved gasses such as O_2, N_2 and CO_2.

Removal of Oxygen:
Partial removal of oxygen can take place during the deaeration as well.

An additional water treatment option is the use of sacrificial anodes for oxygen binding within closed circuits. The magnesium anodes break down magnesium, which oxidizes with oxygen, thus binding the oxygen. The resulting magnesium $Mg(OH)_2$- sludge has to be discharged appropriately.

12.3 Methods of water purification (conditioning)

Water conditioning involves the stabilising of water hardness, the minimising of corrosion and the prevention of water contamination using appropriate additives. The use of chemical additives is relevant for setting the guide values or as measure to limit damages from the unavoidable intrusion of outside water and air.

It is important to keep in mind that conditioning agents may increase the electrical conductivity and influence material compatibility and the purity of used chemicals. In addition, it has to be ensured that the conditioning agents are compatible. The use of conditioning agents must follow hygienic – toxicological standards as well as environmental regulations.

Increase of the pH-Value:
An increase of the pH-value is carried out in order to avoid iron dissolution in the water, which takes place under the formation of iron (II) hydroxide, and to form a protective layer of iron oxides (magnetite) at a flow temperature $\gg$ 100 °C. To reach a suitable protective alkalinity, sodium hydroxide (NaOH) can be used as well as amines, the use of which has improved greatly over the last years.

A pH-value of 9.3 provides the optimum protection condition for iron. Operational experience demonstrates that a pH-value of up to 8.5 ensures failure-free operation. For brass materials the risk of erosion corrosion increases for pH- values > 9 and locally increased flow velocity.

Hardness Stabilization:
The previously described water recycling processes help to set the right degree of hardness. Any unavoidable increase of water hardness caused by outside water intrusion can be controlled by the use of hardness stabilizers and anti-scaling agents. Chemicals based on phosphates, poly-phosphates and poly acrylate have been approved. They help to prevent hard scaling as well as deposits from corrosion products and suspended substances.

Oxygen Reduction:
The infiltration of oxygen is minimal within a self-contained district heating network without increased supplementary water demand in failure-free operation, so that no corrosion damages occur. There is no need for oxygen reduction. Oxygen elimination is necessary if under certain operating conditions an increase in oxygen is expected. The elimination can be conducted by the use of partial flow degasification devices, by catalytic and electro-chemical oxygen elimination and by adding one of the following oxygen binders:

- Hydrazine: Based on its carcinogen rating it should be used only under exceptional conditions. It is prohibited if hot water generation is directly connected.
- Sodium sulphite: Restricted to operating conditions in a saline environment due to the increased salinity. In order to decrease the risk of corrosion the sulphate concentration should not exceed 250 mg/l. It should not be used in combination with copper bearing materials.
- Organic oxygen binders such as ascorbic acid.

Corrosion Inhibitor:
Corrosion inhibitors and their reaction products must neither reduce heat transfer nor trigger corrosion when used correctly. Besides the standard corrosion inhibitors such as phosphates and silicates, mixtures of chemicals are used. These are based on amines, borate, molybdate, nitrite and tannin. All of these agents produce decomposition products such as inorganic and/or organic acids or other organic compositions. Therefore, it is necessary to utilize buffer agents (e.g. phosphates) to keep the pH-Value stable.

13 Specifications Pipe Systems

13.1 Heat transfer capacity at varying temperature differences

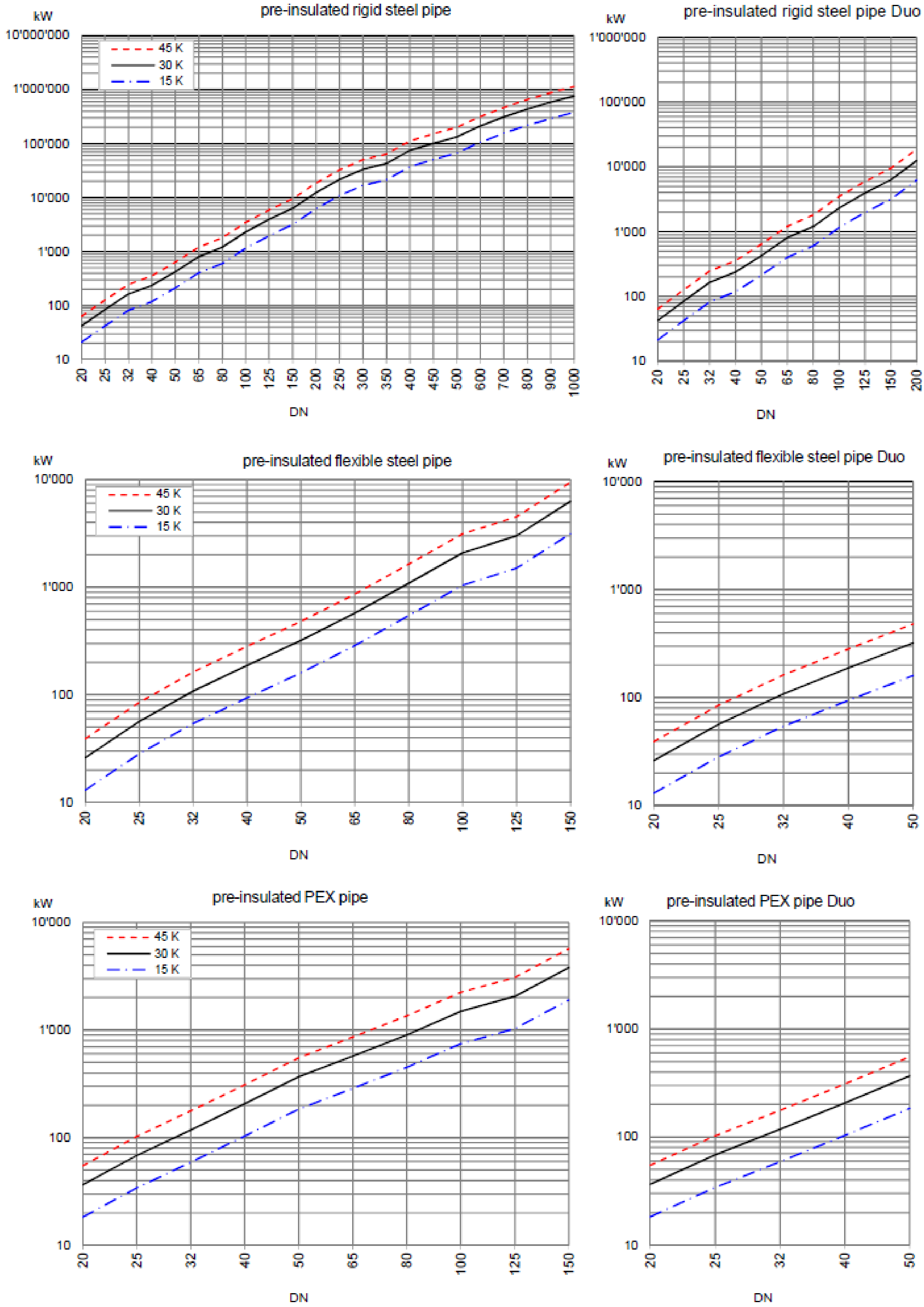


Figure 13.1 Recommended values for heat transfer capacity at temperature differences of 15, 30 and 45 K and a specific pressure loss of 300 Pa/m for the pipe systems pre-insulated rigid steel pipes, pre-insulated flexible steel pipes (corrugated tube) and pre-insulated PEX pipes.

13.2 Heat transfer capacity at varying specific pressure losses

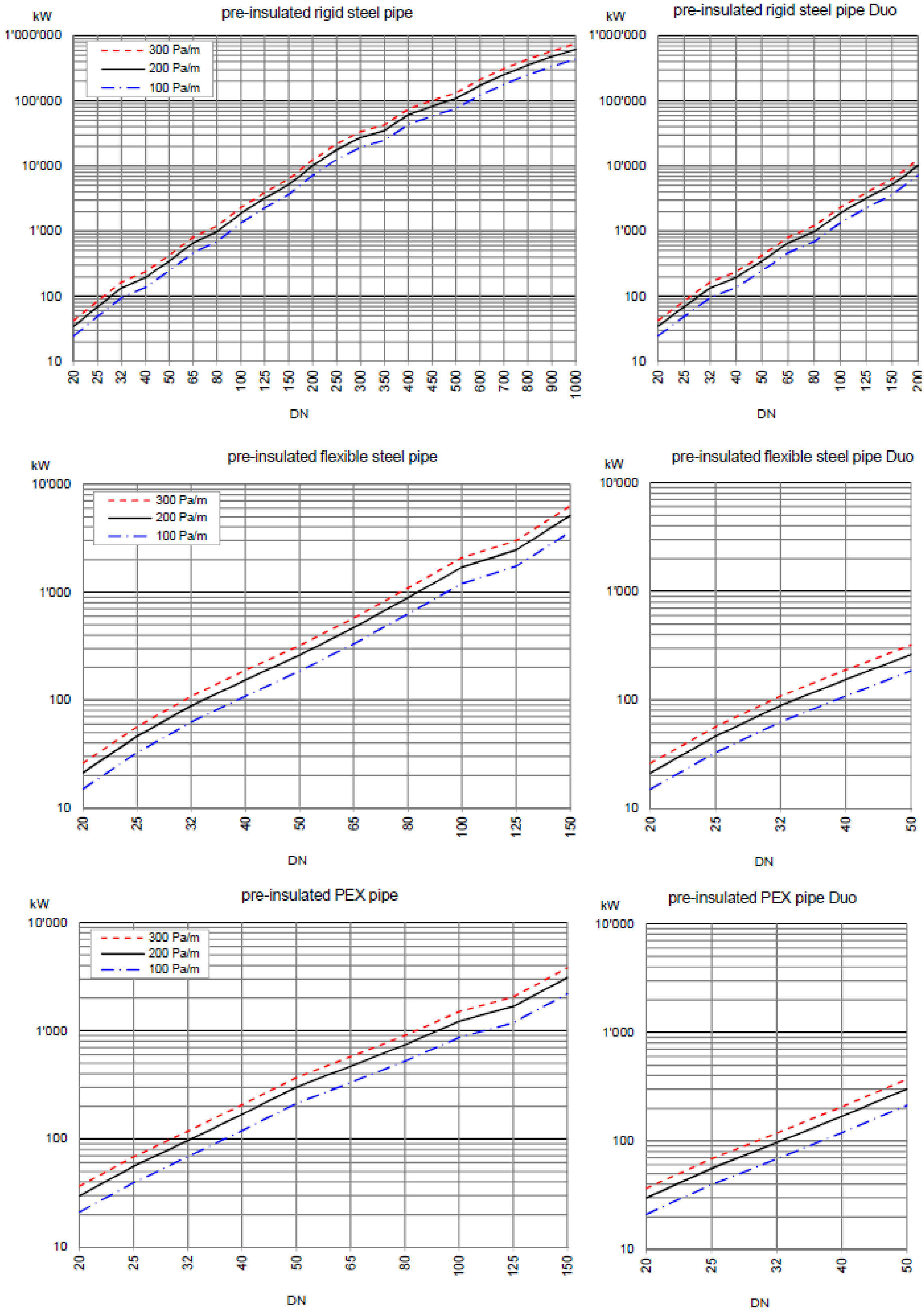


Figure 13.2 Recommended values for heat transfer capacity of specific pressure losses of 100, 200 and 300 Pa/m and a temperature difference of 30 K for the pipe systems pre-insulated rigid steel pipes, pre-insulated flexible steel pipes (corrugated tube) and pre-insulated PEX pipes

13.3 R-Value-sheet for seamless steal pipes

The following data tables are taken from the planning book “QM-Holzheizwerke” [21]. The values represent the specific pressure loss per routed pipeline meter (R -Value). For each R-value, the following additional values were calculated for the nominal diameters DN25, DN32, DN40, DN50, DN65, DN80, DN100, DN125, DN150, DN200 and DN250:

- Mass Flow Rate kg/h (within the table classified as “Massenstrom”)
- Average Flow Velocity m/s (within the table classified as “Geschwindigkeit”)
- Dynamic Pressure Pa (within the table classified as “Staudruck”)

The tables have been calculated with the following constant values:

- Water Temperature 70 °C
- Density 977,7 kg/m^3
- Kinematic Viscosity 4,14 E-7 m^2/s
- Pipe Surface Roughness 0,01 mm
- Pipe Inner Diameter DIN 2448 for seamless Steel Pipes

The specific pressure loss within pipelines per routed meter (R-Value), will be calculated as follows,

$$R = \frac{\lambda}{d_i} \frac{\rho_W}{2} v^2$$

The average flow velocity v will be determined by:

$$v = \sqrt{\frac{d_i}{\lambda} \frac{2}{\rho_W} R}$$

Calculation of the pipe resistance coefficient of the pipe flow (Lambda-Value) follows iterative the Colebrook equation for turbulent flow within a pipe:

$$\frac{1}{\sqrt{\lambda}} = -2\log\left(\frac{k}{d_i\ 3.71} + \frac{2.51}{Re\ \sqrt{\lambda}}\right)$$

Mass Flow Rate and dynamic Pressure relate to the following equation:

$$\dot{m} = \frac{v\ \rho_W\ d_i^2\ \pi}{4}$$

$$p_{dyn} = \frac{v^2\ \rho_W}{2}$$

R [Pa/m]	Nennweite DN [mm]	25	32	40	50	65	80	100	125	150	200	250
	Rohr-Innendurchmesser [mm]	28.5	37.2	43.1	54.5	70.3	82.5	107.1	131.7	159.3	206.5	260.4
10	Massenstrom [kg/h]	303	626	933	1'759	3'490	5'363	10'786	18'731	31'092	61'957	114'561
	Geschwindigkeit [m/s]	0.135	0.164	0.182	0.214	0.255	0.285	0.34	0.391	0.443	0.526	0.611
	Staudruck [Pa]	8.9	13	16	22	32	40	57	75	96	135	183
12	Massenstrom [kg/h]	336	694	1'034	1'948	3'864	5'936	11'930	20'708	34'360	68'436	126'485
	Geschwindigkeit [m/s]	0.15	0.181	0.201	0.237	0.283	0.315	0.376	0.432	0.49	0.581	0.675
	Staudruck [Pa]	11	16	20	28	39	49	69	91	117	165	223
14	Massenstrom [kg/h]	367	757	1'128	2'124	4'211	6'466	12'989	22'539	37'385	74'428	137'511
	Geschwindigkeit [m/s]	0.164	0.198	0.22	0.259	0.308	0.344	0.41	0.47	0.533	0.631	0.734
	Staudruck [Pa]	13	19	24	33	46	58	82	108	139	195	263
16	Massenstrom [kg/h]	396	817	1'216	2'289	4'535	6'962	13'980	24'251	40'214	80'033	147'821
	Geschwindigkeit [m/s]	0.177	0.213	0.237	0.279	0.332	0.37	0.441	0.506	0.573	0.679	0.789
	Staudruck [Pa]	15	22	27	38	54	67	95	125	161	225	304
18	Massenstrom [kg/h]	424	873	1'299	2'445	4'842	7'431	14'916	25'868	42'884	85'319	157'540
	Geschwindigkeit [m/s]	0.189	0.228	0.253	0.298	0.354	0.395	0.47	0.539	0.611	0.724	0.84
	Staudruck [Pa]	17	25	31	43	61	76	108	142	183	256	345
20	Massenstrom [kg/h]	450	926	1'378	2'593	5'133	7'877	15'805	27'403	45'419	90'336	166'764
	Geschwindigkeit [m/s]	0.2	0.242	0.268	0.316	0.376	0.419	0.498	0.572	0.647	0.766	0.89
	Staudruck [Pa]	20	29	35	49	69	86	121	160	205	287	387
22	Massenstrom [kg/h]	475	977	1'454	2'734	5'412	8'302	16'654	28'868	47'839	95'124	175'563
	Geschwindigkeit [m/s]	0.212	0.255	0.283	0.333	0.396	0.441	0.525	0.602	0.682	0.807	0.937
	Staudruck [Pa]	22	32	39	54	77	95	135	177	227	318	429
24	Massenstrom [kg/h]	499	1'026	1'527	2'870	5'679	8'711	17'468	30'273	50'157	99'711	183'992
	Geschwindigkeit [m/s]	0.222	0.268	0.297	0.35	0.416	0.463	0.551	0.631	0.715	0.846	0.982
	Staudruck [Pa]	24	35	43	60	84	105	148	195	250	350	471
26	Massenstrom [kg/h]	522	1'074	1'597	3'001	5'936	9'103	18'251	31'624	52'387	104'121	192'095
	Geschwindigkeit [m/s]	0.233	0.281	0.311	0.365	0.434	0.484	0.576	0.66	0.747	0.883	1.02
	Staudruck [Pa]	26	39	47	65	92	114	162	213	273	381	513
28	Massenstrom [kg/h]	545	1'119	1'664	3'127	6'184	9'482	19'007	32'928	54'538	108'375	199'909
	Geschwindigkeit [m/s]	0.243	0.293	0.324	0.381	0.453	0.504	0.599	0.687	0.777	0.919	1.07
	Staudruck [Pa]	29	42	51	71	100	124	176	231	295	413	556
30	Massenstrom [kg/h]	566	1'163	1'730	3'249	6'424	9'849	19'738	34'188	56'618	112'488	207'462
	Geschwindigkeit [m/s]	0.252	0.304	0.337	0.396	0.47	0.523	0.622	0.713	0.807	0.954	1.11
	Staudruck [Pa]	31	45	55	77	108	134	189	249	318	445	599
35	Massenstrom [kg/h]	618	1'268	1'885	3'539	6'994	10'719	21'472	37'179	61'551	122'237	225'363
	Geschwindigkeit [m/s]	0.275	0.332	0.367	0.431	0.512	0.57	0.677	0.775	0.877	1.04	1.2
	Staudruck [Pa]	37	54	66	91	128	159	224	294	376	526	707
40	Massenstrom [kg/h]	666	1'367	2'031	3'811	7'527	11'534	23'094	39'975	66'162	131'346	242'084
	Geschwindigkeit [m/s]	0.297	0.357	0.395	0.464	0.551	0.613	0.728	0.834	0.943	1.11	1.29
	Staudruck [Pa]	43	62	76	105	148	184	259	340	435	607	815
45	Massenstrom [kg/h]	712	1'459	2'168	4'067	8'030	12'302	24'624	42'611	70'508	139'931	257'837
	Geschwindigkeit [m/s]	0.317	0.382	0.422	0.495	0.588	0.654	0.777	0.889	1.01	1.19	1.38
	Staudruck [Pa]	49	71	87	120	169	209	295	386	494	689	925
50	Massenstrom [kg/h]	755	1'548	2'299	4'311	8'509	13'032	26'076	45'113	74'632	148'074	272'775
	Geschwindigkeit [m/s]	0.336	0.405	0.448	0.525	0.623	0.693	0.822	0.941	1.06	1.26	1.46
	Staudruck [Pa]	55	80	98	135	190	235	331	433	553	771	1'035
55	Massenstrom [kg/h]	796	1'632	2'423	4'543	8'965	13'728	27'461	47'499	78'565	155'838	287'015
	Geschwindigkeit [m/s]	0.355	0.427	0.472	0.553	0.656	0.73	0.866	0.991	1.12	1.32	1.53
	Staudruck [Pa]	61	89	109	150	210	260	367	480	613	854	1'146
60	Massenstrom [kg/h]	836	1'713	2'543	4'766	9'402	14'396	28'789	49'786	82'332	163'272	300'649
	Geschwindigkeit [m/s]	0.372	0.448	0.495	0.581	0.688	0.765	0.908	1.04	1.17	1.39	1.6
	Staudruck [Pa]	68	98	120	165	232	286	403	527	673	938	1'258

Figure 13.3 R-value-table for [21] R-values of 10-60 Pa/m

Nennweite DN [mm]		25	32	40	50	65	80	100	125	150	200	250
Rohr-Innendurchmesser [mm]		28.5	37.2	43.1	54.5	70.3	82.5	107.1	131.7	159.3	206.5	260.4
R [Pa/m]												
65	Massenstrom [kg/h]	874	1'791	2'658	4'981	9'823	15'038	30'065	51'983	85'953	170'417	313'748
	Geschwindigkeit [m/s]	0.389	0.468	0.518	0.607	0.719	0.799	0.948	1.08	1.23	1.45	1.67
	Staudruck [Pa]	74	107	131	180	253	312	439	575	734	1'022	1'370
70	Massenstrom [kg/h]	911	1'866	2'769	5'188	10'229	15'657	31'296	54'103	89'443	177'303	326'373
	Geschwindigkeit [m/s]	0.406	0.488	0.539	0.632	0.749	0.832	0.987	1.13	1.28	1.5	1.74
	Staudruck [Pa]	81	116	142	195	274	339	476	622	795	1'106	1'482
75	Massenstrom [kg/h]	947	1'939	2'877	5'388	10'622	16'256	32'486	56'152	92'818	183'959	338'572
	Geschwindigkeit [m/s]	0.422	0.507	0.56	0.656	0.777	0.864	1.02	1.17	1.32	1.56	1.81
	Staudruck [Pa]	87	126	153	211	295	365	513	670	856	1'191	1'595
80	Massenstrom [kg/h]	982	2'009	2'981	5'583	11'002	16'836	33'640	58'137	96'087	190'405	350'386
	Geschwindigkeit [m/s]	0.437	0.525	0.581	0.68	0.805	0.895	1.06	1.21	1.37	1.62	1.87
	Staudruck [Pa]	93	135	165	226	317	391	550	719	917	1'275	1'708
85	Massenstrom [kg/h]	1'016	2'078	3'083	5'771	11'372	17'400	34'759	60'064	99'259	196'662	361'850
	Geschwindigkeit [m/s]	0.452	0.543	0.6	0.703	0.832	0.925	1.1	1.25	1.41	1.67	1.93
	Staudruck [Pa]	100	144	176	242	339	418	587	767	979	1'361	1'822
90	Massenstrom [kg/h]	1'049	2'145	3'181	5'955	11'731	17'948	35'849	61'938	102'345	202'744	372'994
	Geschwindigkeit [m/s]	0.467	0.561	0.62	0.725	0.859	0.954	1.13	1.29	1.46	1.72	1.99
	Staudruck [Pa]	107	154	188	257	360	445	625	816	1'041	1'446	1'936
95	Massenstrom [kg/h]	1'081	2'210	3'277	6'134	12'082	18'482	36'909	63'763	105'349	208'666	383'843
	Geschwindigkeit [m/s]	0.481	0.578	0.638	0.747	0.884	0.982	1.16	1.33	1.5	1.77	2.05
	Staudruck [Pa]	113	163	199	273	382	472	662	865	1'102	1'532	2'050
100	Massenstrom [kg/h]	1'112	2'273	3'371	6'308	12'424	19'003	37'944	65'542	108'279	214'441	394'420
	Geschwindigkeit [m/s]	0.495	0.594	0.656	0.768	0.909	1.01	1.2	1.37	1.54	1.82	2.1
	Staudruck [Pa]	120	173	211	289	404	499	700	913	1'165	1'618	2'164
110	Massenstrom [kg/h]	1'172	2'396	3'552	6'646	13'084	20'009	39'942	68'979	113'935	225'587	414'836
	Geschwindigkeit [m/s]	0.522	0.626	0.692	0.809	0.958	1.06	1.26	1.44	1.62	1.91	2.21
	Staudruck [Pa]	133	192	234	320	448	553	776	1'012	1'290	1'790	2'394
120	Massenstrom [kg/h]	1'230	2'514	3'726	6'969	13'716	20'974	41'856	72'270	119'350	236'257	434'373
	Geschwindigkeit [m/s]	0.548	0.657	0.726	0.849	1	1.11	1.32	1.51	1.7	2	2.32
	Staudruck [Pa]	147	211	257	352	493	607	852	1'111	1'415	1'964	2'625
130	Massenstrom [kg/h]	1'286	2'627	3'893	7'280	14'325	21'900	43'695	75'433	124'553	246'505	453'137
	Geschwindigkeit [m/s]	0.573	0.687	0.758	0.887	1.05	1.16	1.38	1.57	1.78	2.09	2.42
	Staudruck [Pa]	160	230	281	384	537	662	928	1'210	1'541	2'138	2'857
140	Massenstrom [kg/h]	1'340	2'736	4'055	7'580	14'911	22'794	45'468	78'481	129'567	256'381	471'214
	Geschwindigkeit [m/s]	0.597	0.715	0.79	0.923	1.09	1.21	1.43	1.64	1.85	2.17	2.51
	Staudruck [Pa]	174	250	305	417	582	717	1'005	1'310	1'668	2'312	3'089
150	Massenstrom [kg/h]	1'392	2'842	4'211	7'870	15'478	23'658	47'182	81'426	134'412	265'921	488'675
	Geschwindigkeit [m/s]	0.62	0.743	0.82	0.958	1.13	1.26	1.49	1.7	1.92	2.26	2.61
	Staudruck [Pa]	188	270	329	449	627	773	1'082	1'410	1'795	2'488	3'322
160	Massenstrom [kg/h]	1'443	2'944	4'362	8'151	16'028	24'495	48'842	84'279	139'104	275'158	505'581
	Geschwindigkeit [m/s]	0.642	0.77	0.849	0.993	1.17	1.3	1.54	1.76	1.98	2.33	2.7
	Staudruck [Pa]	202	290	353	482	673	829	1'160	1'510	1'922	2'664	3'556
170	Massenstrom [kg/h]	1'492	3'044	4'509	8'423	16'561	25'307	50'453	87'048	143'657	284'120	521'980
	Geschwindigkeit [m/s]	0.664	0.796	0.878	1.03	1.21	1.35	1.59	1.82	2.05	2.41	2.78
	Staudruck [Pa]	216	309	377	514	718	884	1'238	1'611	2'050	2'840	3'791
180	Massenstrom [kg/h]	1'539	3'140	4'652	8'689	17'080	26'097	52'019	89'739	148'082	292'830	537'917
	Geschwindigkeit [m/s]	0.686	0.821	0.906	1.06	1.25	1.39	1.64	1.87	2.11	2.48	2.87
	Staudruck [Pa]	230	329	401	547	764	940	1'316	1'712	2'178	3'017	4'026
190	Massenstrom [kg/h]	1'586	3'235	4'791	8'947	17'586	26'866	53'545	92'359	152'390	301'309	553'429
	Geschwindigkeit [m/s]	0.706	0.846	0.933	1.09	1.29	1.43	1.69	1.93	2.17	2.56	2.95
	Staudruck [Pa]	244	350	425	580	810	997	1'394	1'814	2'307	3'194	4'261

Figure 13.4 R-value-table [21] for R-values of 65-190 Pa/m

Nennweite DN [mm]		25	32	40	50	65	80	100	125	150	200	250
Rohr-Innendurchmesser [mm]		28.5	37.2	43.1	54.5	70.3	82.5	107.1	131.7	159.3	206.5	260.4
R [Pa/m]												
200	Massenstrom [kg/h]	1'631	3'327	4'927	9'200	18'078	27'617	55'032	94'914	156'590	309'574	568'549
	Geschwindigkeit [m/s]	0.727	0.87	0.959	1.12	1.32	1.47	1.74	1.98	2.23	2.63	3.03
	Staudruck [Pa]	258	370	450	614	856	1'053	1'472	1'916	2'436	3'372	4'497
220	Massenstrom [kg/h]	1'719	3'504	5'189	9'687	19'030	29'066	57'903	99'846	164'696	325'524	597'722
	Geschwindigkeit [m/s]	0.766	0.916	1.01	1.18	1.39	1.54	1.83	2.08	2.35	2.76	3.19
	Staudruck [Pa]	287	410	499	680	949	1'167	1'630	2'120	2'695	3'728	4'971
240	Massenstrom [kg/h]	1'803	3'675	5'440	10'153	19'941	30'453	60'651	104'565	172'453	340'783	625'628
	Geschwindigkeit [m/s]	0.803	0.961	1.06	1.24	1.46	1.62	1.91	2.18	2.46	2.89	3.34
	Staudruck [Pa]	315	451	549	748	1'042	1'281	1'789	2'325	2'954	4'086	5'446
260	Massenstrom [kg/h]	1'884	3'839	5'682	10'602	20'817	31'786	63'292	109'099	179'903	355'435	652'420
	Geschwindigkeit [m/s]	0.839	1	1.11	1.29	1.52	1.69	2	2.28	2.56	3.02	3.48
	Staudruck [Pa]	344	492	599	815	1'135	1'395	1'948	2'531	3'215	4'444	5'922
280	Massenstrom [kg/h]	1'962	3'997	5'915	11'034	21'662	33'070	65'836	113'467	187'080	369'548	678'223
	Geschwindigkeit [m/s]	0.874	1.04	1.15	1.34	1.59	1.76	2.08	2.37	2.67	3.13	3.62
	Staudruck [Pa]	373	534	649	883	1'229	1'510	2'107	2'738	3'477	4'804	6'400
300	Massenstrom [kg/h]	2'038	4'149	6'140	11'452	22'477	34'311	68'294	117'686	194'012	383'177	703'139
	Geschwindigkeit [m/s]	0.908	1.08	1.2	1.39	1.65	1.82	2.15	2.45	2.77	3.25	3.75
	Staudruck [Pa]	403	575	699	951	1'323	1'626	2'268	2'945	3'739	5'165	6'879
350	Massenstrom [kg/h]	2'217	4'512	6'674	12'443	24'410	37'251	74'114	127'673	210'417	415'427	762'085
	Geschwindigkeit [m/s]	0.987	1.18	1.3	1.52	1.79	1.98	2.34	2.66	3	3.52	4.07
	Staudruck [Pa]	477	680	826	1'123	1'561	1'916	2'671	3'466	4'398	6'071	8'080
400	Massenstrom [kg/h]	2'385	4'850	7'173	13'367	26'214	39'995	79'543	136'989	225'716	445'494	817'029
	Geschwindigkeit [m/s]	1.06	1.27	1.4	1.63	1.92	2.13	2.51	2.86	3.22	3.78	4.36
	Staudruck [Pa]	551	786	954	1'296	1'800	2'209	3'076	3'990	5'061	6'982	9'287
450	Massenstrom [kg/h]	2'542	5'169	7'643	14'238	27'912	42'577	84'653	145'754	240'107	473'771	868'692
	Geschwindigkeit [m/s]	1.13	1.35	1.49	1.73	2.04	2.26	2.67	3.04	3.42	4.02	4.63
	Staudruck [Pa]	627	893	1'083	1'470	2'041	2'503	3'484	4'517	5'727	7'897	10'499
500	Massenstrom [kg/h]	2'692	5'471	8'089	15'064	29'522	45'025	89'493	154'056	253'737	500'545	917'603
	Geschwindigkeit [m/s]	1.2	1.43	1.58	1.83	2.16	2.39	2.82	3.21	3.62	4.25	4.9
	Staudruck [Pa]	703	1'000	1'213	1'645	2'283	2'799	3'894	5'047	6'396	8'814	11'715
550	Massenstrom [kg/h]	2'835	5'760	8'514	15'851	31'056	47'356	94'104	161'962	266'715	526'035	964'162
	Geschwindigkeit [m/s]	1.26	1.51	1.66	1.93	2.27	2.52	2.97	3.38	3.8	4.46	5.14
	Staudruck [Pa]	779	1'108	1'344	1'822	2'526	3'097	4'306	5'578	7'067	9'735	12'933
600	Massenstrom [kg/h]	2'972	6'036	8'920	16'604	32'524	49'587	98'514	169'524	279'127	550'410	1'008'679
	Geschwindigkeit [m/s]	1.32	1.58	1.74	2.02	2.38	2.64	3.11	3.54	3.98	4.67	5.38
	Staudruck [Pa]	856	1'217	1'475	1'999	2'771	3'396	4'719	6'111	7'740	10'658	14'155
650	Massenstrom [kg/h]	3'103	6'301	9'311	17'328	33'934	51'730	102'749	176'784	291'042	573'806	1'051'402
	Geschwindigkeit [m/s]	1.38	1.65	1.81	2.11	2.48	2.75	3.24	3.69	4.15	4.87	5.61
	Staudruck [Pa]	934	1'326	1'607	2'177	3'016	3'695	5'133	6'645	8'414	11'583	15'380
700	Massenstrom [kg/h]	3'230	6'557	9'688	18'025	35'292	53'793	106'828	183'776	302'515	596'332	1'092'534
	Geschwindigkeit [m/s]	1.44	1.71	1.89	2.2	2.58	2.86	3.37	3.83	4.31	5.06	5.83
	Staudruck [Pa]	1'012	1'436	1'740	2'356	3'262	3'996	5'549	7'181	9'091	12'511	16'607
750	Massenstrom [kg/h]	3'353	6'804	10'052	18'699	36'603	55'787	110'767	190'527	313'592	618'078	1'132'239
	Geschwindigkeit [m/s]	1.49	1.78	1.96	2.28	2.68	2.96	3.49	3.97	4.47	5.24	6.04
	Staudruck [Pa]	1'090	1'547	1'873	2'535	3'509	4'298	5'965	7'719	9'769	13'440	17'836
800	Massenstrom [kg/h]	3'471	7'044	10'404	19'351	37'874	57'716	114'579	197'061	324'312	639'122	1'170'658
	Geschwindigkeit [m/s]	1.55	1.84	2.03	2.36	2.77	3.07	3.61	4.11	4.62	5.42	6.25
	Staudruck [Pa]	1'168	1'657	2'007	2'715	3'757	4'600	6'383	8'257	10'448	14'370	19'067
850	Massenstrom [kg/h]	3'587	7'276	10'746	19'984	39'106	59'588	118'277	203'398	334'708	659'527	1'207'908
	Geschwindigkeit [m/s]	1.6	1.9	2.09	2.43	2.86	3.17	3.73	4.24	4.77	5.59	6.44
	Staudruck [Pa]	1'247	1'768	2'141	2'896	4'005	4'903	6'802	8'797	11'129	15'303	20'299

Figure 13.5 R-value-table [21] for R-values of 200-850 Pa/m

13.4 Dimensions and specific heat losses for pre-insulated rigid, flexible steel and PEX Pipes

Table 13.1 **Pre-insulated rigid steel pipes:** Dimensions and specific heat loss for the nominal sizes from DN20 - DN1000. Data from the following manufacturers were used: Brugg Pipe Systems, Isoplus and Logstor

Nominal Width	Medium-Inner Pipe				Outer Diameter Jacket Tube (Thermal Insulation Class)			Specific Heat Loss per routed Meter		
	Outer Diameter	Wall Thickness	Inner Diameter	Volume Inner Pipe	TIC1	TIC2	TIC3	TIC1	TIC2	TIC3
DN	mm	mm	Mm	l/m	mm	mm	mm	W/(m K)	W/(m K)	W/(m K)
20	26.9	2.65	21.60	0.37	90	110	125	0.284	0.248	0.229
25	33.7	2.60	28.50	0.64	90	110	125	0.342	0.291	0.266
32	42.4	2.60	37.20	1.09	110	125	140	0.354	0.317	0.290
40	48.3	2.60	43.10	1.46	110	125	140	0.403	0.356	0.322
50	60.3	2.90	54.50	2.33	125	140	160	0.450	0.398	0.350
65	76.1	2.90	70.30	3.88	140	160	180	0.527	0.446	0.393
80	88.9	3.20	82.50	5.35	160	180	200	0.547	0.469	0.416
100	114.3	3.60	107.10	9.01	200	225	250	0.576	0.490	0.432
125	139.7	3.60	132.50	13.79	225	250	280	0.663	0.562	0.482
150	168.3	4.00	160.30	20.18	250	280	315	0.777	0.633	0.531
200	219.1	4.50	210.10	34.67	315	355	400	0.844	0.670	0.555
250	273.0	5.00	263.00	54.33	400	450	500	0.820	0.656	0.556
300	323.9	5.60	312.70	76.80	450	500	580	0.933	0.744	0.578
350	355.6	5.60	344.40	93.16	500	560	630	0.912	0.719	0.589
400	406.4	6.30	393.80	121.80	560	630	730	0.964	0.744	0.579
450	457.2	6.30	444.60	155.25	630	670	800	0.970	0.839	0.605
500	508.0	6.30	495.40	192.75	710	800	900	0.941	0.728	0.595
600	610.0	7.10	595.80	278.80	800	900	1000	1.125	0.836	0.679
700	711.0	8.00	695.00	379.37	900	1000	1100	1.266	0.938	0.761
800	813.0	8.80	795.40	496.89	1000	1100	1200	1.409	1.042	0.842
900	914.0	10.00	894.00	627.72	1100	1200	–	1.542	1.141	–
1000	1016.0	11.00	994.00	776.00	1200	1300	–	1.678	1.241	–

Table 13.2 **Pre-insulated rigid steel pipe duo:** Dimensions and specific heat loss for the nominal sizes from DN20 – DN200. Data from the following manufacturers were used: Brugg Pipe Systems, Isoplus and Logstor

Nominal Width	Medium-Inner Pipe				Outer Diameter Jacket Tube (Thermal Insulation Class)			Specific Heat Loss per routed Meter		
	Outer Diameter	Wall Thickness	Inner Diameter	Volume Inner Pipe	TIC1	TIC2	TIC3	TIC1	TIC2	TIC3
DN	mm	mm	mm	l/m	mm	mm	mm	W/(m K)	W/(m K)	W/(m K)
20	26.9	2.60	21.70	0.37	125	140	–	0.204	0.184	–
25	33.7	2.60	28.50	0.64	140	160	–	0.223	0.195	–
32	42.4	2.60	37.20	1.09	160	180	–	0.242	0.213	–
40	48.3	2.60	43.10	1.46	160	180	–	0.286	0.243	–
50	60.3	2.90	54.50	2.33	200	225	–	0.280	0.241	–
65	76.1	2.90	70.30	3.88	225	250	–	0.329	0.278	–
80	88.9	3.20	82.50	5.35	250	280	–	0.371	0.297	–
100	114.3	3.60	107.10	9.01	315	355	–	0.375	0.300	–
125	139.7	3.60	132.50	13.79	400	450	–	0.363	0.293	–
150	168.3	4.00	160.30	20.18	450	500	–	0.419	0.330	–
200	219.1	4.50	210.10	34.67	560	630	–	0.475	0.349	–

Table 13.3 **Pre-insulated flexible steel pipe:** Dimensions and specific heat loss for the nominal sizes from DN20 – DN150. Data from the following manufacturers were used: Brugg Pipe Systems.

Nominal Width	Medium-Inner Pipe				Outer Diameter Jacket Tube (Thermal Insulation Class)			Specific Heat Loss per routed Meter		
	Outer Di-ameter	Wall Thick-ness	Inner Di-ameter	Volume Inner Pipe	TIC1	TIC2	TIC3	TIC1	TIC2	TIC3
DN	mm	mm	mm	l/m	mm	mm	mm	W/(m K)	W/(m K)	W/(m K)
20	25.5	0.3	22.00	0.38	91	–	–	0.245	–	–
25	34.0	0.3	30.00	0.71	91	111	–	0.307	0.265	–
32	43.8	0.4	38.90	1.19	111	126	–	0.325	0.294	–
40	54.5	0.5	48.50	1.85	111	126	–	0.401	0.354	–
50	66.5	0.5	60.00	2.83	126	142	–	0.443	0.390	–
65	85.6	0.60	75.80	4.51	178	–	–	0.396	–	–
80	109.2	0.80	98.00	7.54	178	233	–	0.542	0.394	–
100	142.9	0.90	127.00	12.67	233	–	–	0.540	–	–
125	162.7	1.00	147.00	16.97	233	–	–	0.683	–	–
150	218.0	1.20	197.50	30.64	313	–	–	0.693	–	–

Table 13.4 **Pre-insulated flexible steel pipe Duo:** Dimensions and specific heat loss for the nominal sizes from DN20 – DN50. Data from the following manufacturers were used: Brugg Pipe Systems.

Nominal Width	Medium-Inner Pipe				Outer Diameter Jacket Tube (Thermal Insulation Class)			Specific Heat Loss per routed Meter		
	Outer Di-ameter	Wall Thick-ness	Inner Di-ameter	Volume Inner Pipe	TIC1	TIC2	TIC3	TIC1	TIC2	TIC3
DN	mm	mm	mm	l/m	mm	mm	mm	W/(m K)	W/(m K)	W/(m K)
20	25.5	0.3	22.00	0.38	111	–	–	0.156	–	–
25	34.0	0.3	30.00	0.71	126	–	–	0.181	–	–
32	43.8	0.4	38.90	1.19	142	–	–	0.224	–	–
40	54.5	0.5	48.50	1.85	162	–	–	0.251	–	–
50	66.5	0.5	60.00	2.83	182	225	–	0.293	0.215	–

Table 13.5 **Flexible pre-insulated PEX Pipes:** Dimensions and specific heat loss for the nominal sizes from DN20 – DN150. Data from the following manufacturers were used: Brugg Pipe Systems and Isoplus.

Nominal Width	Medium-Inner Pipe				Outer Diameter Jacket Tube (Thermal Insulation Class)			Specific Heat Loss per routed Meter		
	Outer Diameter	Wall Thickness	Inner Diameter	Volume Inner Pipe	TIC1	TIC2	TIC3	TIC1	TIC2	TIC3
DN	mm	mm	mm	l/m	mm	mm	mm	W/(m K)	W/(m K)	W/(m K)
20	25.0	2.30	20.40	0.33	75	90	–	0.264	0.235	–
25	32.0	2.90	26.20	0.54	75	90	–	0.321	0.279	–
32	40.0	3.70	32.60	0.83	90	110	–	0.332	0.284	–
40	50.0	4.60	40.80	1.31	110	125	–	0.341	0.307	–
50	63.0	5.80	51.40	2.07	125	140	–	0.378	0.340	–
65	75.0	6.80	61.40	2.96	140	160	–	0.405	0.356	–
80	90.0	8.20	73.60	4.25	160	180	–	0.429	0.380	–
100	110.0	10.00	90.00	6.36	160	180	–	0.557	0.476	–
125	125.0	11.40	102.20	8.20	180	–	–	0.567	–	–
150	160.0	14.60	130.80	13.44	250	–	–	0.511	–	–

Table 13.6 **Flexible pre-insulated PEX Pipe Duo:** Dimensions and specific heat loss for the nominal sizes from DN20 – DN50. Data from the following manufacturers were used Brugg Pipe Systems and Isoplus.

Nominal Width	Medium-Inner Pipe				Outer Diameter Jacket Tube (Thermal Insulation Class)			Specific Heat Loss per routed Meter		
	Outer Diameter	Wall Thickness	Inner Diameter	Volume Inner Pipe	TIC1	TIC2	TIC3	TIC1	TIC2	TIC3
DN	mm	mm	mm	l/m	mm	mm	mm	W/(m K)	W/(m K)	W/(m K)
20	25.0	2.30	20.40	0.33	90	110	–	0.211	0.174	–
25	32.0	2.90	26.20	0.54	110	125	–	0.215	0.198	–
32	40.0	3.70	32.60	0.83	125	140	–	0.235	0.222	–
40	50.0	4.60	40.80	1.31	160	180	–	0.264	0.210	–
50	63.0	5.80	51.40	2.07	180	–	–	0.246	–	–

13.5 Specific heat loss per meter pipeline

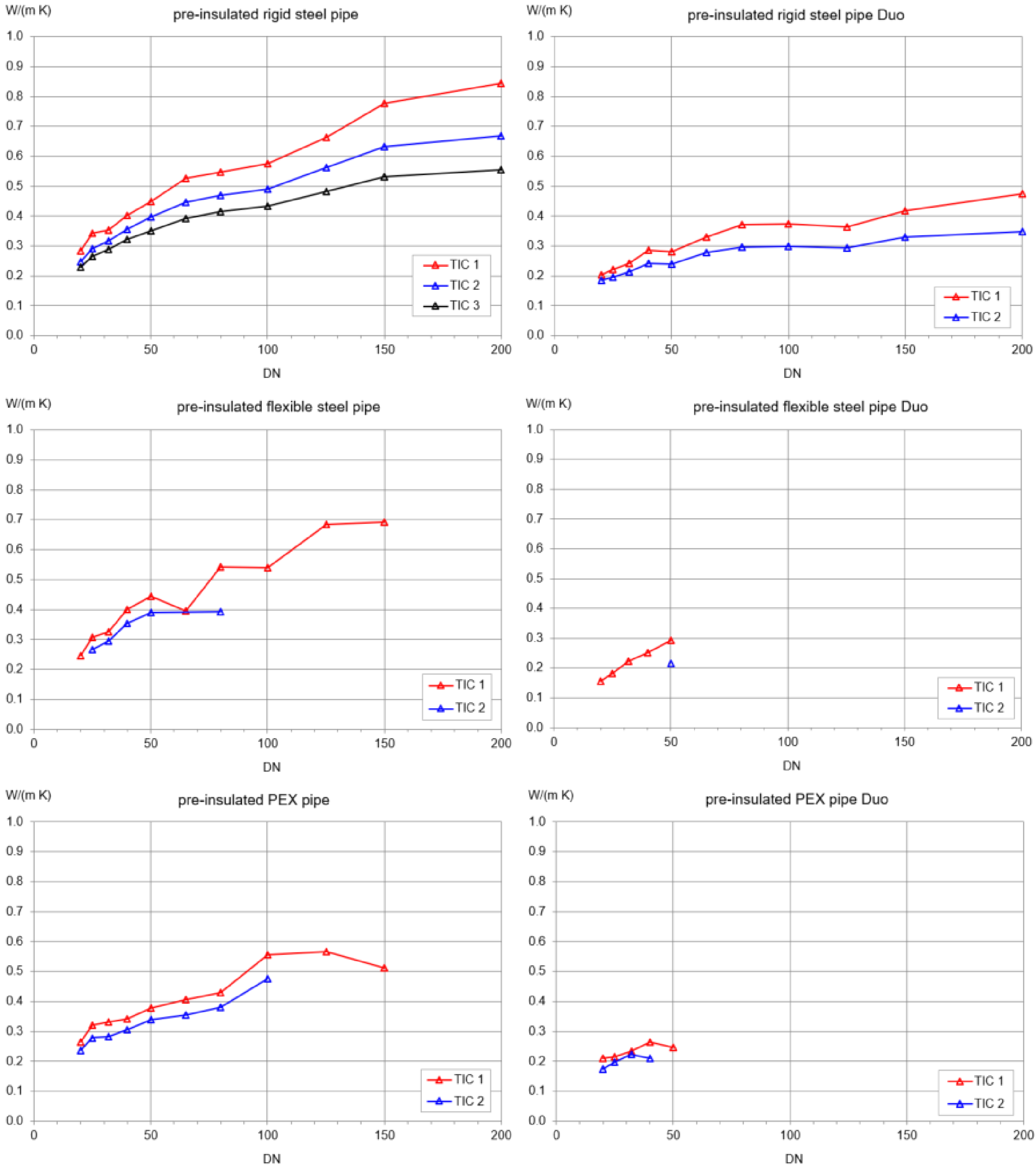


Figure 13.6 Specific heat loss per routed meter pipeline until DN200 for pre-insulated rigid steel pipes, pre-insulated flexible steel pipes, and pre-insulated PEX pipes. On the right side are the standard double specifications. Single pipes dimensions are calculated following the specifications described in chapter 7.1.3. Double pipe specifications from Brugg Pipe Systems, Isoplus and Logstor
The calculations for single pipes rely on the specifications:
- Heat Conductivity Soil λ_{Soil} = 1.2 W/(m K)
- Heat Conductivity Insulation Agent λ_{IM} = 0.03 W/(m K)
- Medium Depth of Cover h_{Cov} = 0.6 m
- Clearance Pipe Distance a = 0.2 m.

13.6 Recommended prices for district heating pipes

The following information applies to recommended prices charged in Switzerland (as of 2009). This information should be treated with caution, especially when applied in other countries and currencies.

Figure 13.7 shows indicative prices of the costs for pre-insulated rigid pipes including laying and civil engineering as summarized in Table 13.7 and Table 13.8. Civil engineering is additionally divided into costs for trench digging in open terrain without paved surface and laying under roads (paved surfaces, pavement), which must be restored after completion of the work.

The additional data points shown in Figure 13.7 represent implemented district heating networks in Switzerland and are taken from an actual state analysis of district heating networks from 2012 (published in [16] in 2014). The data points represent the investment costs for the district heating distribution network per route metre as a function of the average internal diameter of the district heating pipes used. The average inner pipe diameter was calculated using a function based on 134 district heating networks in Sweden [3]. The data points in Figure 13.7 come from district heating networks that were built almost exclusively with pre-insulated rigid pipes and were commissioned between 1984 and 2011.

Since prices for material and labour are constantly changing and the planning process between preliminary study and implementation of district heating networks can take several years, any cost adjustments during implementation must be taken into account. Seasonal differences are also possible, especially for civil engineering work. Compared to favourable conditions, specific situations can also lead to increased costs, especially due to the high complexity of the pipe routing (e.g. cobblestones, crossing of rivers, motorways, railway tracks).

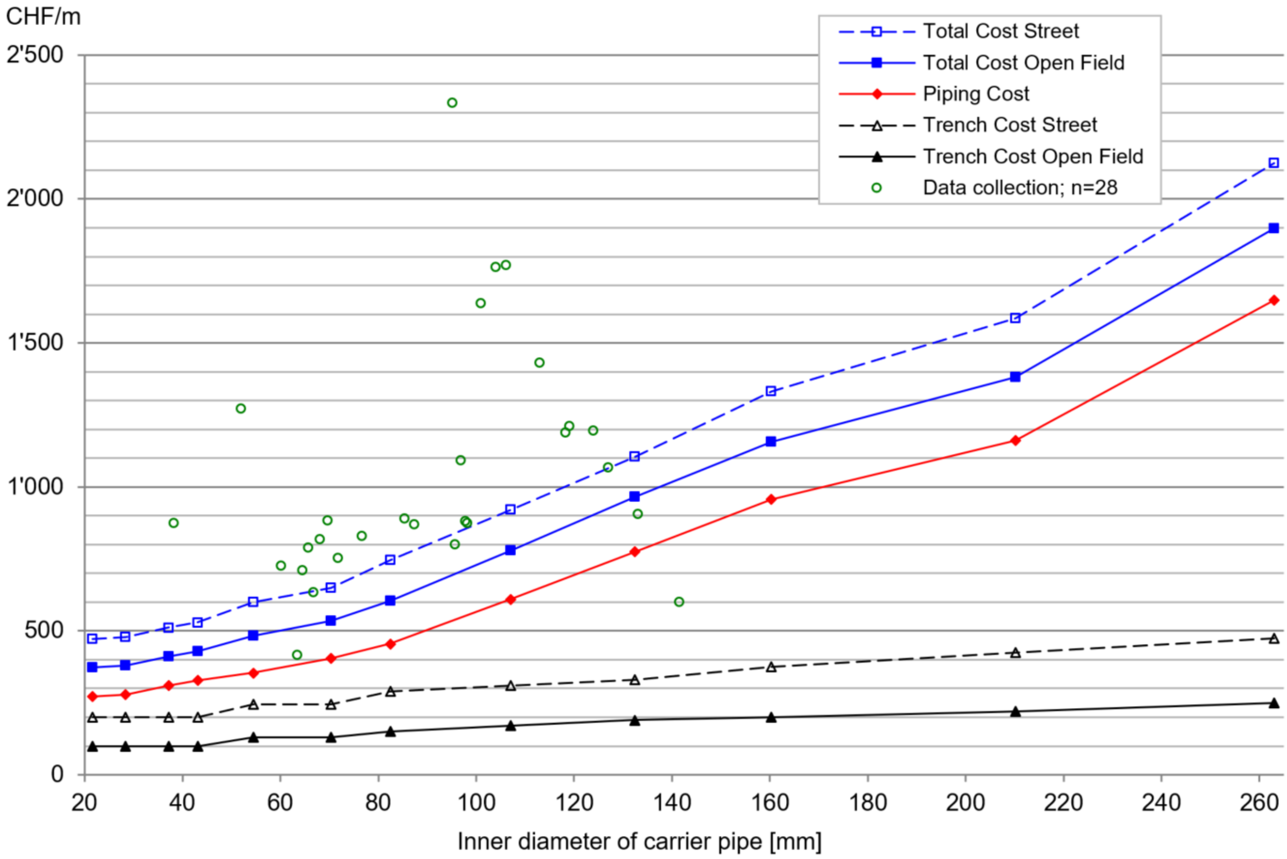


Figure 13.7 Recommended costs for pre-insulated rigid steel pipes (2009) from DN20 - DN250 divided into pipe and trench digging costs for laying in the fields and street. In addition, data from existing district heating networks from a data collection of 2012 [16] are listed.

Table 13.7 Compilation of the included services for the recommended prices for district heating pipelines shown in Table 13.8 and .

Material and Piping	Trench Excavation	Comments
- Supply- and Return per single Meter - 1/10 Bend Prize Share - 5 % T-Piece Individual Share - 1/6 Socket Costs - 10 % for Wall Sealing Rings, Expansion Zones, Pipe Bedding and Welding Material - 1/100 Price Share for Monitoring Devices, Pipe laying and welding of the Pipes - Pressure Test	- Excavated Material (Based on Documents) - 30 % drive off of Excavation Material - Pipe Sand Bedding - Sanding of Pipes - Backfill of trenches - Restoration of the earth surface - Road Surfacing	DN20 – DN150 Thermal Insulation Class 3 DN200 – DN250 Thermal Insulation Class 2 This price does not include: - Weld Seam inspection by digital radiography - Allocation of Plant Pipelines - Trench Shoring - Traffic Control

Table 13.8 Recommended prices for district heating pipelines (2009). Values shown in italics for pre-insulated rigid steel pipes from DN 300 are extrapolated values. If necessary, the prices should be specifically requested from the manufacturer, as the use of pipe dimensions from DN 300 upwards is rarely used, especially in Switzerland.

Nominal Width	Pre-insulated rigid steel pipe		Pre insulated rigid steel pipe Duo		Pre-insulated flexible steel pipe		Pre-insulated flexible steel pipe Duo		Pre-insulated PEX pipe		Pre-insulated PEX pipe Duo	
	Parcel	Street	Parcel	Street	Parcel	Street	Parcel	Street	Parcel	Street	Parcel	Street
DN	CHF/m	CHF/m	CHF/m	CHF/m	CHF/m	CHF/m	CHF/m	CHF/m	CHF/m	CHF/m	CHF/m	CHF/m
20	373	473	295	374	350	450	252	352	246	346	193	278
25	379	479	299	378	363	463	275	375	258	358	222	322
32	411	511	325	404	453	553	309	409	306	406	245	350
40	429	529	339	418	465	585	348	448	323	423	315	415
50	484	599	382	473	524	634	385	485	458	548	352	452
65	535	650	423	514	726	821	–	–	508	618	–	–
80	605	745	478	589	809	904	–	–	613	708	–	–
100	780	920	616	727	917	1052	–	–	667	762	–	–
125	965	1105	762	873	1027	1162	–	–	721	816	–	–
150	1157	1332	914	1052	1106	1241	–	–	775	870	–	–
200	1381	1586	1091	1253	–	–	–	–	–	–	–	–
250	1899	2124	–	–	–	–	–	–	–	–	–	–
300	*2417*	*2662*	–	–	–	–	–	–	–	–	–	–
350	*2935*	*3200*	–	–	–	–	–	–	–	–	–	–
400	*3453*	*3738*	–	–	–	–	–	–	–	–	–	–
450	*3971*	*4276*	–	–	–	–	–	–	–	–	–	–
500	*4489*	*4814*	–	–	–	–	–	–	–	–	–	–
600	*5007*	*5352*	–	–	–	–	–	–	–	–	–	–
700	*5525*	*5890*	–	–	–	–	–	–	–	–	–	–
800	*6043*	*6428*	–	–	–	–	–	–	–	–	–	–
900	*6561*	*6966*	–	–	–	–	–	–	–	–	–	–
1000	*7079*	*7504*	–	–	–	–	–	–	–	–	–	–

14 Questionnaire for a district heating connection

The questionnaire can be seen on the following two pages.

Questionnaire for a District Heating Connection

Contact Data ☐ Owner ☐ Administration ☐ Tenant

Name: ....

First Name: ....

Street: .... House Number: ....

Address Supplemental: ....

Zip Code, City: ....

Email: .... Phone. ....

Object Data Street: .... House Number: ....

Address Supplemental, Zip Code City: ....

Building Type: ☐ SFH Description (separated, attached, Town House, etc.): ....

☐ MFH ☐ Commercial building Number of floors: ....

☐ Residential and Commercial Building Number of apartments: ....

☐ Several Buildings with a Heat Transfer Station. Number of buildings: ....

☐ Industrial Operation with Process Heat Description: ....

☐ .... Description: ....

Type of Use: ☐ Residential Number of persons: ....

☐ Commercial Residential area: .... %

☐ .... Description use: ....

Year of construction ....

Energy reference area .... m^2 ERA

Heat Generation Boiler type: .... Year: ....

(Recent Condition) Boiler output: .... kW (info found on boiler)

More than one heat generator: ☐ Yes ☐ No

Description ....

Remarks ....

Energy Demand Energy demand within the past three years for space heating, domestic hot water and process heat

Heat period				
Fuel	l/a			
Gas	m^3/a			
Wood Pellets	t/a			
Wood Chips	Srm/a			
Split Logs	Cord/a			
Electricity	kWh/a			

☐ Exact (e.g. Delivery Note)

☐ Estimation

☐ Excluded domestic hot water

☐ Included domestic hot water

Ratio of domestic hot water: .... %

Remarks: ....

Energy demand for cooling: ☐ Yes ☐ No

Description ....

Questionnaire for a District Heating Connection

Heat Distribution / Heat Supply

Number of heating groups: ....

Description: ....

Heat transfer system
- ☐ Floor heating — Max. temperatures supply/return .... / .... °C
- ☐ Radiator — Max. temperatures supply/return .... / .... °C
- ☐ Heating register — Max. temperatures supply/return .... / .... °C
- ☐ Ventilation facility
  - ☐ with heat regulation — Max. temperatures supply/return .... / .... °C
  - ☐ without heat regulation

Remarks ....

Domestic hot water generation with district heating: ☐ Yes ☐ No

If yes, domestic hot water generation occurs: ☐ throughout the whole year ☐ only seasonally

Number of Residents: ....

Type:
- ☐ Instantaneous
- ☐ Storage tank with integrated heat exchanger
- ☐ Storage tank with external heat exchanger
- ☐ ....

Volume storage tank: .... Litre

Integrated circulation pump: ☐ Yes ☐ No

Remarks ....

Restoration Intended energy relevant restoration that are not included in the prior stated energy demand.

	Realization (Year):	Impact on the energy demand (reduction)
☐ Window	....	.... %
☐ Insulation outer wall	....	.... %
☐ Insulation roof	....	.... %
☐ Solar energy for domestic hot water generation	....	.... %
☐ Solar energy for domestic hot water generation and heat supply	....	.... %
☐	....	.... %
☐	....	.... %

Remarks: ....

Connection date ☐ immediately ☐ intermediate term (until 5 Years) ☐ long-term up to 10 Years ☐ No Interest

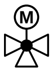
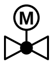
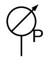
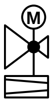
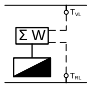
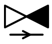
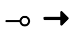
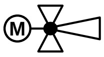
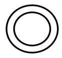
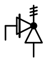
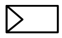
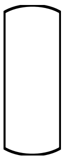
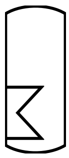
Possible connection date .... (Year)

Attachments
- ☐ Location plan heat station (possible location transfer station)
- ☐ Location plan/ sketch for Initiating a possible house connection
- ☐ ....
- ☐ ....
- ☐ ....

15 Installation Symbols

The following table lists the symbols used within this handbook. They are based on the SIA 410 [85].

Table 15.1 Symbols according to the Swiss Engineers and Architects Association SIA [85]

Symbol	Explanation	Symbol	Explanation
	Motor driven Three-Way valve		Strainer / dirt trap
	Motor driven Straight-Way valve		Pressure indicator
	Throttle valve with pressure measuring connection		Temperature indicator
	Pressure independent control valve (PICV)		Heat meter
	Check valve		Sensor (temperature and pressure)
	Shut-off valve		Pump
	Jet pump with actuator		Heating circuit
	Spring-loaded pressure relief valve		Controller
	Draining		Heat exchanger (eg. plate heat exchanger)
	Hot water storage tank		Thermal energie storage tank with integrated heat exchanger

16 Formula Symbols and Indices

Symbol	Explanation	Unit
A	area, surface	m^2
a	annuity factor	%/a
a	distance corrugated tube	m
a	inner tube clearance	m
c	concurrency, coincidence factor	–
C	costs	CHF
c	specific costs	CHF/MWh
C	circumference	m
C_I	investment cost	CHF
c_P	specific heat capacity	kJ/(kg K); J/(kg K)
d	discount rate	m
d	diameter	m
DN	nominal diameter	–
E	energy	MWh, kWh
E	elastic modulus	N/mm^2
ERA	energy reference area	m^2
f	factor	–
f_{el}	weighting factor electricity	–
g	gravitational acceleration	m/s^2
H	delivery head	m; mWS
h	elevation, depth	m
i	capital interest rate	%/a
k	pipe surface roughness	mm
k_V	flow characteristic valve	$m^3/s \cdot \sqrt{bar}$
k_{VS}	flow characteristic open valve	$m^3/s \cdot \sqrt{bar}$
L	length of pipeline (supply and return line in parallel)	m
L	length of the main pipe	m
l	length	m
L_A	leg length	m
LHD	linear heat density	MWh/(a m)
m	mass	kg
$\dot{m}$	mass flow rate	kg/s
$\dot{M}$	mass flow rate expansion	kg/s
n	amount	–
n	calculation run time	a
N	valve authority	–
P	power (usually electrical)	kW

Symbol	Explanation	Unit
P	Price (eg. fuel price)	CHF/MWh
p	pressure	bar, Pa, kPa
Δp	pressure difference, pressure drop	bar, Pa, kPa
Δp_{V0}	pressure drop valve at a flow rate of 0% during valve opening	bar, Pa, kPa
Δp_{V100}	pressure drop valve at a flow rate of 100%	bar, Pa, kPa
Δp_{var100}	Pressure drop across the line with variable flow rate	bar, Pa, kPa
PLS	pipeline section	–
PS	maximum operating pressure	bar
Q	annual heat capacity / Heat quantity	MWh/a, kWh/a / kWh
$\dot{Q}$	thermal capacity	MW, kW
$\dot{q}$	specific thermal capacity	W/m
q_h	specific heat demand	MJ/(m² a), kWh/(m² a)
q_{HW}	specific heat demand for hot water generation	MJ/(m² a), kWh/(m² a)
r	radius	m
Re	reynolds number	–
S	safety factor	–
s	horizontal tube clearance	m
s	wall thickness (pipe, cylinder)	m
s_e	ordered wall thickness	m
S_{UTS}	safety factor tensile strength	–
s_V	calculated wall thickness	m
S_{YTS}	safety factor yield strength	–
T	torgue	Nm; Nmm
T	temperature	°C
ΔT	temperature difference	K
t	time	h
TIC	thermal insulation class	–
U	Overall heat transfer coefficient	W/(m² K)
UTS	ultimate tensile strength	N/mm²
V	volume	m³, m³/a
$\dot{V}$	volumetric flow rate	m³/s
ΔV	volume change (expansion volume)	m³, m³/a
v	flow velocity	m/s
W	section modulus	N/mm³
YTS	yield strength	N/mm²
α	heat transfer coefficient	W/(m² K)
α	thermal expansion coefficient	1/K
γ	volume change coefficient	1/K

Symbol	Explanation	Unit
ζ	coefficient of resistance of installations	–
η	efficiency	%
η_a	annual fuel use efficiency	%
λ	friction factor	–
λ	thermal conductivity, thermal resistance	W/(m K)
ν	kinematic viscosity	m^2/s
ν_N	weld seam quality	–
π	circle number, pi,	–
ρ	density	kg/m^3
σ	tension	N/mm^2
τ	full load operating hours	h/a

Indices	Explanation
0	begin, origin, input
1	end, output
a	year, annual
a	ambient, outside
acc	acceptable
AMB	ambient, environment
AS	actual state
B	bending
Bldg	building
c	concurrency, coincidence factor
C	Capital
CN	critical node
cool	cooling
Cov	vertical cover
DES	design
Duo	two-pipe system, double pipeline
EC	excess consumption
el	electric
el	electricity (costs)
ELP	earth-laid pipeline
ex	exergy
fuel	fuel
gr	ground (soil)
HC	heat consumer / house connection
HD	heat distribution

Indices	Explanation
HGR	heat generator (heat station)
HM	heat meter
HWT	storage (heat water storage tank)
hydr	hydraulic
i	inner
i	numbering of pipeline section or house connection
I	investment
IM	insulation material
JP	jacket pipe
K	boiler (kettle)
L	Leg length
L	linear
L	loss
Lay	laying
low	lower
m	mean
maint.	maintenance and service work
max	maximum
min	minimum
MuW	Make-up Water
N	nominal
N	network
NPV	net present value
OP	operation
opt	optimal

Indices	Explanation
OR	operating resources
P	pump
P	pipe, medium pipe
PBS	pressure boosting station
PLS	pipeline section
PM	pressure maintenance
PN	nominal pressure
Pre	prestress
PSD	pipe static design
q	heat
REF	reference
RTN	return flow
S/R	supply- and return flow
SPLY	supply flow
sply	supplied
St	static
T	temperature
TES	Thermal Energy Storage
tot	total
UH	usable heat
up	upper
V	equivalent tensile stress
W	water
WC	water column
weighted	weighted
x	searched variable

17 Glossary

Term	Explanation
ambient heat, environ-mental heat	Ambient heat or environmental heat is a renewable, natural and widely available form of energy at a relatively low tem-perature level. Sources of ambient heat are the air, the upper soil as well as groundwater, sea and river water. Heat pumps can be used to raise ambient heat to a higher temperature level and make it usable. This requires the supply of high-quality energy usually in the form of electricity from another source.
annual fuel use effi-ciency	Utilization degree within a year. Find furthermore information under the chapter 2.6.1 Efficiency and utilization degree.
annual heat demand	The annual heat demand of a consumer equals the heat demand at the point of heat supply. For a district heating network, the annual heating requirement represents the annual heat demand, at the interface between heat generation and the heat distribution network.
annual duration curve of the outside temperature	The annual load duration curve of the outside temperature is a representation of the sum frequency of the ambient tem-perature as number of days or hours per year for a particular measuring station. It reflects the cumulating frequency curve of the outside temperature.
annual load duration curve of the heat load demand	The annual load duration curve of the heat load demand consists out of the load curve and the annual load duration curve of the ambient temperature. It is a cumulating frequency curve that represents the heat load demand in relation to the annual amount of days or hours.
annual operation period	The effective annual operating hours of a facility. The annual operating hours will not be represented by full utilization hours in other words one operating hour with an efficiency rate of 50 % equals one hour of operation.
annual performance fac-tor, annual COP	The annual utilization degree describes the ratio between the annual heat production and the annual supplied electrical energy for operating a heat pump. Find additional information under the coefficient of performances.
districts and zones	A potential heat supply area may include a village, quarters, neighborhoods, several key customers or one major cus-tomer. The potential heat supply area is divided into areas and zones due to the expected heat demand density of building types or due to geographical conditions such as roads, railway lines, streams etc. Individual zones can be treated simply as major customers. A local map or an energy register assists in the classification of a village.
base load/ minimum load	Base load denotes a permanent capacity that is required within 8760 hours per year. The base load of a district heating network consists out of the seasonal independent consumers plus network losses during base load operation.
base load	When two heat generators will be utilized then the heat load demand is divided into a base- and a peak load component. A high number of operating hours are assigned to the base load generator whereas a peak load boiler will be assigned to a lower number.
base load coverage	Heat generation unit to cover the base load.
basement pipeline	The basement pipeline connects the house connection pipeline with the transfer station.
biomass	Biomass covers all plant and animal substances. In the context of energy technology, all biogenic substances are gener-ally considered as energy sources. Wood and fermentable waste are primarily used for district heating.
boiler efficiency	The boiler efficiency consists out of the useful upstream produced energy by a boiler divided by the calorific value of the fuel as supplied energy. The determination takes place either in the stationary state without storage effects (e.g. in the case of automated combustion) or via a combustion process (e.g. in the case of hand firing).
branch pipes	Branches or distribution lines departure from main lines to individual group of consumers.
building types	Categorization of buildings by design, type of use, year of construction or additional criteria. Examples of these classifica-tions are single- and multifamily houses, old- and new buildings, residential areas and industrial zones as well as the distances among buildings.
central heating	Central heating is used for heat supply of a building by a central heat generator.
coefficient of perfor-mance (COP)	The performance factor describes the ratio between the heats generated over an extended time period by a heat pump to the provided electrical energy within this observation period. Find additional information under annual operation period and performance coefficient.
coefficient of perfor-mance (cop)	The coefficient of performance of a heat pump can be described by the ratio generated heat output to supply energy. It describes an instantaneous value or a value based on a short-term observation.
cold water	Cold drinking water whose temperature has not been intent ally increased.
combined heat and power (CHP)	A facility for combined heat and power (CHP) utilizes thermal generated by a combustion motor or a steam turbine as power generator for example. It includes the generation of electricity and at the same time the utilization of waste heat in a thermal process as useful heat. Combined heat and power plants will be also classified as block heat and power plants. Thermal power plants with waste heat utilization will be labeled as combined heat and power plant or cogeneration plants.

Term	Explanation
combined heat and power plant (CHP plant), cogeneration plant	Energy center with simultaneous cogeneration of heat and electricity. See also combined heat and power (CHP).
concurrency, concurrency factor	Within a network of heat consumers, the concurrency describes the effect that by increasing numbers of consumers never all of them will be supplied at the same time with the maximum rated heat capacity. The coincidence factor equals 1 for a single heat consumer and it becomes smaller than 1 for multiple heat consumers. It describes the ratio between the effectively expected maximum loss performances of all heat consumers to the total subscribed connected load of the heat consumers.
connected annual heat demand ratio	The degree of connection describes the ratio between the annual heat demands of the connected consumers within an area related to the annual heat demand of all eligible consumers within this area. For areas with similar consumer types it defines also the proportion of the amount of the connected heat consumers.
connection load	The connection load of a district heating network is the sum of the connected loads of all heat consumers under consideration of simultaneity or the product that consists out of the sum of all subscribed connection services of all heat consumers and the coincidence factor. Find additional information under concurrency and coincidence factor.
consumer	See also heat consumer.
consumer composition	The consumer composition describes a supply area according to criteria such as building density and development structure linear heat density, demand, concurrency, etc.
consumer's installation	The consumer's installation is the link between the transfer station and domestic heat distribution. It is used to adjust the domestic heat distribution in terms of pressure, temperature and volumetric flow rate. In the design of the consumer's installation, a distinction must be drawn between direct or indirect connection.
critical node	Location of the lowest differential pressure between supply and return. This place can move around within the network in relation to the actual heat demand. The critical node serves as interpretation size for the main pump facility.
degree of utilization, utilization degree	The degree of utilization is the ratio between the generated energy over an extended period of consideration to the energy supplied during this consideration period. This corresponds to the ratio between the effective capacities that add up during the observation period (e.g. the summed up generated heat measured by a heat meter) divided by the summed up supplied capacity within this observation period (e.g. the heating value of the combusted fuel). When the consideration takes place over a period of one year, it is called the degree of annual utilization. If the relation of useful- to supplied energy will be assessed over a brief observation period or as momentary value it will be classified as efficiency degree (see also efficiency).
distribution line	See also branch pipes.
districts and area	A potential heat supply area may include a village, quarters, neighborhoods, several key customers or one major customer. The potential heat supply area is divided into districts and areas due to the expected heat demand density of building types or due to geographical conditions such as roads, railway lines, streams etc. Individual areas can be treated simply as major customers. A local map or an energy register assists in the classification of a village.
district heating	If the heat generation of a district heating network is carried out with a heating system, it is also called district heating.
district heating network	A district heating network is a pipeline system with all the required equipment to supply customers with heat. Water or steam serves as heat carriers. The carrier flows from the heat generator to the heat consumers and returns in a closed system. The closed system constitutes the district heating network.
district heating plant	A district heating plant is a heat station for supplying a district heating network.
district heating local heat, micro grid	District heating describes a conduct-bound heat supply to customers via water and steam with centrally generated heat. District heating networks cover a wide range of power with connected loads of less than 100 kW up to more than 1 GW. For the total energy statistic of the federal government it is also assumed that the main transport and distribution network uses public land and that the heat is sold to third parties [11]. Extended heat networks within a legal entity, for example a major development, are technical identical to a district heating network, but are not recorded as district heating. For smaller networks the term local heat will be used. In Germany this describes the heat transfer for heating and hot water between buildings with a capacity of between 50 kW and a few megawatts [6]. Minergie® utilizes local heat even if the heat production plant supplies some buildings or building complexes, although it does not necessarily have to be sold to third parties [7]. Within this handbook the term district heating rules out of the fact that the transition between local – and district heat is fluent.
domestic heat distribution	The system consists of the distribution system in the building for the distribution of space- and process heating as well as domestic hot water.
domestic hot water	See hot water.
double pipeline execution	Supply and return pipe with PUR foam as thermal insulation in a plastic pipe used as protection. Rigid and flexible versions with pre-insulated steel or pre-insulated PEX pipes are available.

Term	Explanation
drinking water	According to Swiss food legislation, drinking water is defined as naturally left or destined for treatment for drinking, cooking, preparing food and cleaning items that come into contact with food [67]. Drinking water is also used for personal care and purification (shower and bath water, etc.).
earth-laid pipelines	Earth-laying of district heating pipes in a canal, a ditch or trenchless.
efficiency	The efficiency of a technical system describes the relationship between useful- and supplied energy. Under stationary conditions without distortion due to storage effects, the efficiency can also be determined as a relationship between useful capacity and supplied capacity. Within this handbook, the concept of efficiency is used for a moment value determined by the service or a value that relies to a short observation period. In order to rate a plant operation over an extended period of observation, the degree of utilization describes the ratio between the amount of useful capacity added up within the observation period to the amount of supplied capacity during this consideration period (find additional information underutilization- ration or degree).
final capacity stage	The predicted final capacity stage of the district heating network for design and calculation issues.
full-load operating hours, fully operational number of hours	The full-operating number is the annual energy demand divided by the rated heat output. It is an important characteristic of plant dimensioning for a single consumer within the entire system. For example, one full-operation hour equals one hour of operation at nominal load or two hours of operations at 50% load and it applies the rule that the number of full-load operating hours ≤ to the number of annual operating hours.
geodetic network profile, elevation curve	The geodetic mesh profile represents the height variation of the net in meters above sea level the vertical datum used within Switzerland.
geographic information system (GIS)	Data processing application to collect, edit, organize, analyze and present spatial data. In order to plan a district heating networks, routing can be determined taking into account the geographical conditions and additional supply system that may already exist (water, gas, electricity, etc.). In addition to this, the GIS can also be used to estimate local energy and performance needs.
heat carrier medium	The medium used for heat transfer in the heat distribution network, such as water, steam or thermal oil.
heat consumer	Receives heat from the heat supplier and pays for the heat according to the contractual conditions.
heat demand density	The heat demand density is the annual heat demand of all buildings of the supplied area in relation to the base area.
heat exchanger	A heat exchanger is an apparatus in which thermal energy is transferred from a warm mass flux to another colder mass flux.
heat generation plant	A heat generator converts final energy into utility heat and transfers this to a heating medium.
heat load density	Maximum heat output related to an area.
heating plant, district heating plant	Power house for heat generation.
heat supplier	Delivers the contractually agreed heat supply to the heat consumer.
heat supply contract	In the heat supply contract, the interface between the heat supplier and the heat consumer is contractually agreed. The heat supply contract usually contains the following components of the contract: General terms and conditions, technical connection regulations (TCR) and a tariff sheet.
high temperature water	High temperature water in the district heating refers to the circulation water in the district heating network so that the temperature exceeds 110 °C.
hot water	The term hot water is used differently in building service engineering and district heating as follows: In district heating technology, hot water describes the circulation water in the district heating network when the temperature is up to 110 °C while circulation water rated above 110 °C is called high temperature water. Hot water in the district heating network does not have to have drinking water quality and is therefore not to be confused with hot water utilized the building technology. In the building technology hot water stands for heated drinking water that is heated if necessary or made available in hot water storage facilities at temperature of around 60 °C. In order to distinguish heated drinking water from circulatory water in the district heating network, it is referred in this handbook as domestic hot water, provided by a domestic water heating facility.
house connection pipeline	Interconnector between heat distribution network and transfer station.
house substation	The house substation consists out of the transfer station and the consumer's installation.
house substation room	The house substation room included the house substation and the main shut-off valve.
hydraulic network separation	The network separation refers to the separation of wo hydraulic networks, for example by a heat exchanger or a hydraulic separator. Grid separations result in additional energy losses, both in heat and flow energy.
key customer	A key customer is a customer (or a potential customer in the planning phase) with extended heat consumption within the area that will be assessed.

Term	Explanation
least temperature difference (LTD)	The least or terminal temperature difference (LTD) describes the difference between primary and the secondary return temperature at the heat exchanger of transfer station. It is a measure of the quality of heat transfer and should be as small as possible.
length of pipeline	Length of route of main, branch and house connection pipelines. Out of the condition that there will be one pipe for supply and one for return the pipeline length doubles the length of the route.
linear heat density	The linear heat density describes the heat delivered per year and routed meter to the heat consumer. It serves as a measure for assessing energy density of the heat distribution and influences the energy efficiency and the economy of the network. The linear heat density can be presented for the entire network and for partial lines.
load characteristic curve	Presentation of the heat load demand depending on the average daily ambient temperature.
local heat	See district heating.
main pipe	The main line from the heat station to the branch lines within the heat distribution network, usually without house connections. Other terms include e.g. main distribution or transport line if the heat station is actually remote located from the supply area.
main pipeline	See main pipe.
maximum admissible working temperature	Maximum operating temperature allowed over a short period of time.
maximum pressure	Pressure that should never be exceeded at any point or time within the network.
maximum temperature continuous operation	Maximum operating temperature allowed without time restriction.
mesh network	A mesh network or mesh net is a network that joins lines or rings together at several junctions.
micro grids	See district heating
minimum pressure	Pressure rate that cannot be undercut at any point within the net at any time.
network capacity, maximum network capacity, actual network capacity	The maximum network capacity is the predefined required heat output at the entrance of the district heating network. It results from the heat load demand of all consumers multiplied by the concurrency factor plus the heat distribution losses. The current network capacity during regular operation corresponds to the current heat load demand of all customers, but it can also be limited by the current heat generation power in the event of a malfunction, for example.
network pressure difference	The network pressure difference refers to the pressure difference between the supply and return of the entire district heating network.
network separation	Network separation refers to the separation of two hydraulic networks, for example by a heat exchanger or a hydraulic separator. Grid separations lead to additional energy losses, both in terms of heat and flow energy.
network temperatures	Network temperatures refer by a common indication of the supply and return temperature in degrees Celsius (e.g. (80/50).
nominal diameter (DN)	Reference diameter that define the size and compatibility of the components. The nominal diameter is part of the name of the component that relates to EN ISO 6708 and is not identical to the numerical value in millimeters.
nominal heat output capacity	Maximum continuous performance output of a plant which it is designed for according the manufacturers specifications without any restrictions in time.
nominal pressure (PN)	The nominal pressure indicates a reference size for the pipe system. The information relates to DIN, EN or ISO and is provided by the classification PN (Pressure nominal) followed by a number indicating the design pressure in bar for a room temperature of (20 °C9 described by EN 1333.
operating water	Operating water for commercial and domestic applications that does not need to have drinking water quality.
peak load	Maximum required capacity that is decisive for design and relates to the ambient temperature.
peak load coverage	Heat generation facility to cover peak load demands. It should be assigned to an extended regulating array and should be capable of immediate switch on and off. As additional redundancy, the peak load boiler is often designed to compensate the failure of one or more base load boilers.
pipe coefficient of friction pipe-friction coefficient	Dimensionless metric to calculate the pressure drop of a flow within the pipe.
pipe static calculation	Calculation methods for assessing resistance and design pipelines and pipe components.
pipe surface roughness	Characteristic of a surface (in this case the inside tube wall) that classifies the roughness in millimeters.
power density	Maximum performance related to an area. For district heating the density of heat capacity is of interest.
pre-fabricated duct technology	Prefabricated or locally produced concrete channels to embed district heating lines.

Term	Explanation
pre-insulated flexible steel pipe	Flexible steel medium pipe with PUR foam as thermal insulation and a plastic jacket tube as protection. The steel medium pipe is often designed as corrugated tube.
pre-insulated PEX pipe	Flexible plastic medium pipe with PUR foam as thermal insulation and a plastic jacket as tube protection.
pre-insulated rigid steel pipe	Rigid steel media pipe with PUR foam as thermal insulation and a plastic jacket as tube protection.
pressure and expansion maintenance	Subsystem in the hydraulic system (heat generation and heat distribution), which absorbs the volume change of hot water between minimum and maximum temperature and thus keeps the pressure widely constant (pressure maintenance).
pressure pattern diagram	Presentation of the pressure distribution within the grid depending on the distance from the heat station.
primary return temperature	Temperature of the district heating medium, with flows from the heat consumer to the heat generator.
primary side	The primary side is the part of the system flooded by the district heating medium.
primary supply temperature	Temperature of the district heating medium, which flows from the heat generator to the heat consumer.
redundancy	Provision of an additional functional unit, which is not needed during regular operation, in order to increase operational safety.
ring network	In the case of a ring network, one or more lines are merged into one ring. This can increase security of supply.
route, pipeline	The route describes the terrain area required to manage the district heating line.
seasonal operation	Seasonal delivery and supply of heat to the heat consumer within the heating season and the transition period.
secondary return temperature	Temperature of the heating water coming back from individual consumers to the transfer station.
secondary side	The secondary side is the part of the system that will be flow through by the heating medium of the domestic heat distribution system.
secondary supply temperature	Temperature of the heating water from the transfer station to the individual consumers. The heat transfer station at the consumer's side is called secondary because it is usually a hydraulic system separated from the district heating network.
situation assessment	Situation assessment is an analysis of the actual situation with the measurement of energy and capacity demand for heat (space heating, hot water and process heat), the structural situation of the routing and the potential heat supply area.
star network	The network lines initiate from a supply point star-shaped. They will be supplied by one side only.
steel jacket pipe	Rigid steel medium pipe with a steel jacket as protection. Thermal insulation is generally provided by the vacuum between medium and jacket pipe.
subscribed connection load	Contractually agreed maximum heat supply capacity for a consumer connected to a district heating network.
surface roughness	See pipe surface roughness
system pressure	The system pressure describes the pressure within a district heating network.
tariff sheet	The tariff sheet is part of the heat supply contract and regulates the conditions for providing the heat supply.
technical connection requirements (TCR)	The technical connections requirements (or technical connection standards) ideally regulate all technical relevant connection conditions such as pressure, temperature, material, measuring equipment, billing etc. These apply to the planning, connection and operation of the district heating network. The technical connection requirements are part of the heat supply contract.
temperature difference	Temperature difference between supply and return temperature. In the case of a district heating network, the temperature difference of the primary side is usually of interest, i.e. in the district heating network. Within this handbook the temperature difference will be presented in Kelvin.
thermal energy storage (TES)	Thermal energy storage for district heating networks is often operated by unpressurized tanks filled with water. They are used to balance the heat demand of the district heating network by covering load peaks through stored capacities that will be recharged during periods of low heat demand. This allows for a smaller sizing and optimised operation during heat generation. The size of the storage depends on the size of the heat generation system and the task of the storage facility. The capacity ranges from a few cubic meters up to several thousand cubic meters.
thermal insulation classes	The thermal insulation thickness denotes the class of thermal insulation around the medium pipe. For pre-insulated rigid steel pipes three classes are offered with 1 describing the weakest and 3 describing the best insulation. Two classes of insulation are provided for the pre-insulated flexible steel pipes and pre-insulated PEX pipes. They are classified as **s**tandard- and **r**einforced version.

Term	Explanation
transfer station	The transfer station is the link between the house connection pipeline and the consumer's installation. It serves to transfer heat and measure heat supply in accordance with the contract.
trench technique	In the field excavation technology, the district heating lines are buried in direct contact to the soil in open trenches. It is the most common laying process.
trenchless pipeline construction	During the trenchless pipeline construction, the pipes will be laid by inserting, pulling, pipe jacking or pipe ramming into a created cavity within the soil.
utility pipes	The term includes sewer, water, sewage and power lines from municipality, a city or a company.
waste heat	Unavoidable heat loss from energy conversion plants or chemical processes. Waste heat generated during a conducted process can be transferred to another process. The usable potential relies to the amount of heat released over the year and the temperature level of the waste heat. For district heating, waste heat sources can be utilized directly at temperatures above 70 °C with a high number of full-time operating hours.
year-round operation	Year-round delivery and supply of heat to heat consumers.

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